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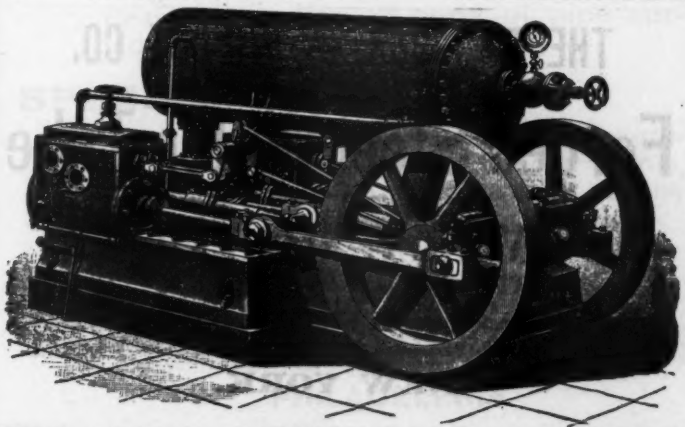

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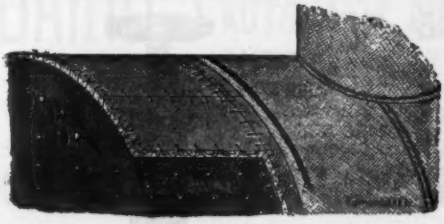

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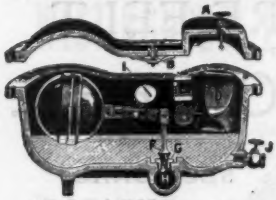

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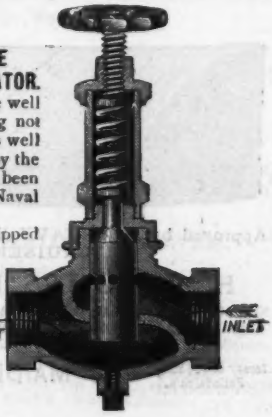

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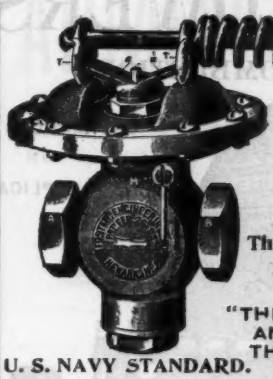

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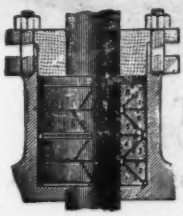

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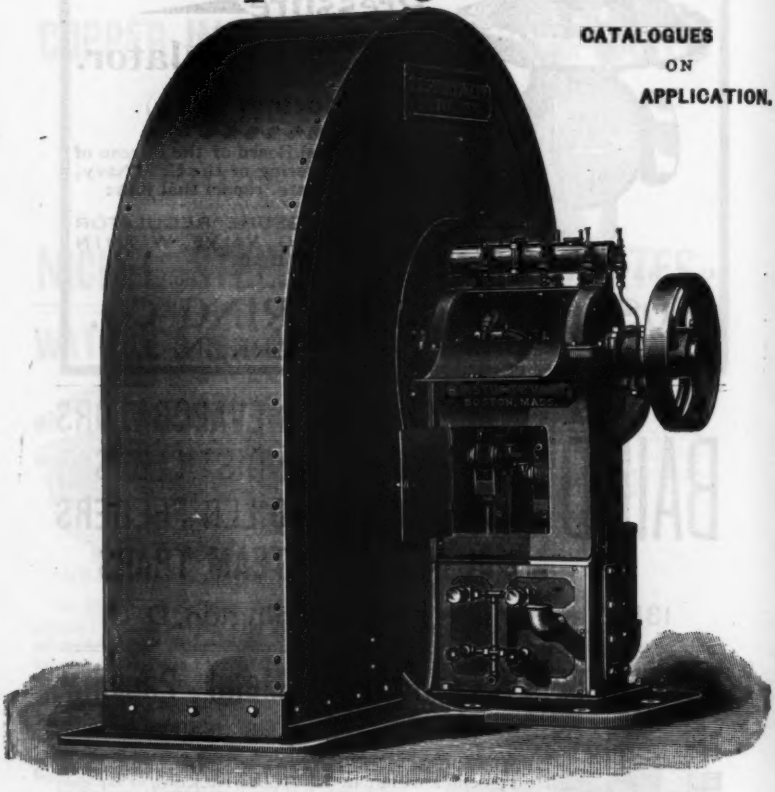

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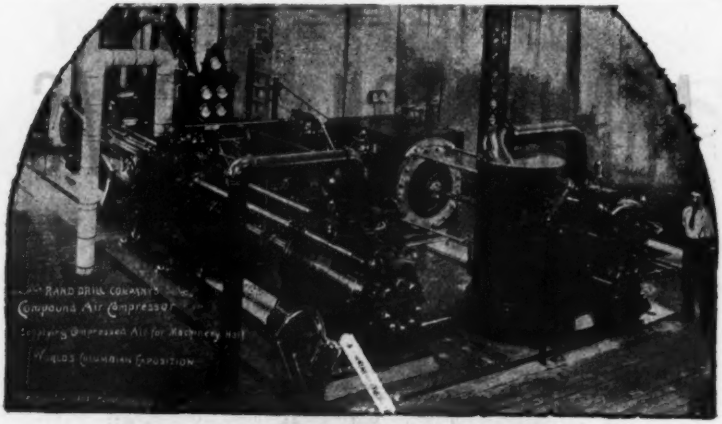

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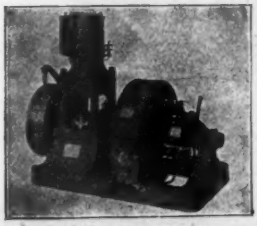

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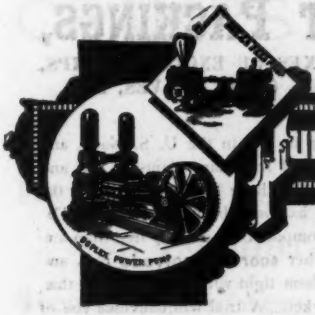

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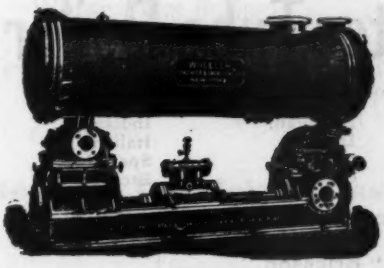

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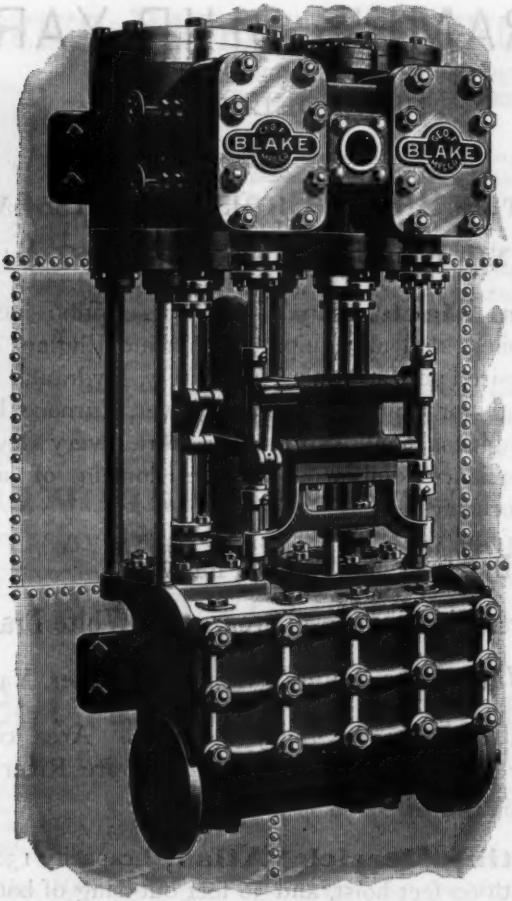

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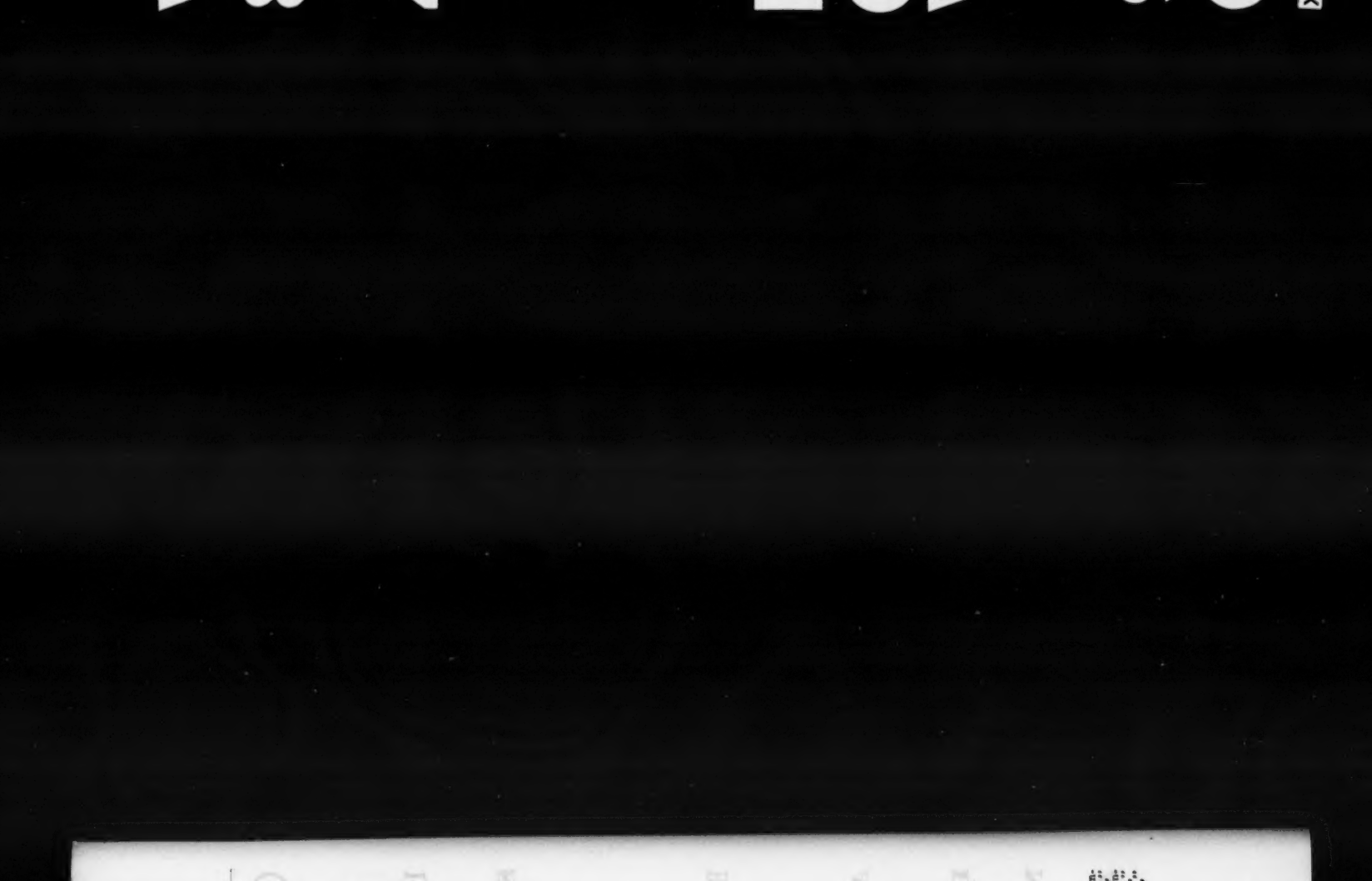

NOTICE.

The Annual Meeting for the Election of Officers for 1897, and the transaction of other business, will be held at the Navy Department, Saturday, January 9, 1897. Members who cannot be present are requested to send proxies.

Members and subscribers are respectfully notified that the annual subscription of $5 for the JOURNAL for the year 1897 is now due, and that prompt payment will be a favor to the Secretary.

It is requested that any change in the address of members, subscribers and exchanges for the coming year be communicated to the Secretary and Treasurer before January 1, 1897.

Respectfully,

F. C. BIEG,
Secretary and Treasurer.

JOURNAL

OF THE

AMERICAN SOCIETY OF NAVAL ENGINEERS.

VOL. VIII. NOVEMBER, 1896. No. 4.

The Society as a body is not responsible for statements made by individual members.

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ON CRANK SHAFTS.

BY JOHN H. MACALPINE, ESQ., ASSOCIATE MEMBER.

The following investigation, which has occupied my time in the Bureau of Steam Engineerimg at intervals during the last two years, deals with the estimation of the various stresses which are induced in the crank shaft of an engine when running. The original object of the investigation, and the principal practical result, is the ascertaining of the dimensions of a set of crank shafts which, under working conditions, will not in any part have stresses rising above a definite value. So far only the shafts with three cranks at 120° have been treated. The results are represented by curves from which can readily be deduced the dimensions of the various parts of the shaft in any particular case.

Usually the diameter of the crank shaft has been determined from the forces due to steam alone, and this is proper for those engines which do not run faster than a moderate speed. But for a large proportion of marine engines built now the inertia forces must be taken into account if a reasonably accurate calculation of the stresses to which the shaft is subjected is to be made.

In what follows there are therefore two separate solutions of the crank shaft problem given:—

1st. For forces due to steam alone.

2d. For forces due both to steam and inertia.

In both cases the constants used have been deduced from the actual practice in a number of engines of the U. S. Navy, which have been successfully tried.

In estimating the forces acting on the shaft it is allowable to suppose that the engine has been carefully built and is intelligently run. To suppose a large error of judgment in either case would be as irrational as to allow for a very serious flaw which might be hidden in the material. We must, then, take the engine running at full power, and may suppose a fairly equal distribution of power among the three cranks: the main bearings and other parts being in reasonably good adjustment.

Determining the constants from successful practice is more or less equivalent to making an allowance for any abnormal stress which is as likely to occur in one engine as in another. In other words, one function of the factor of safety is to cover unusual conditions which we cannot foresee.

When these suppositions are made, the principal factors in producing the stresses in the shaft are reduced to a small number. They are:—

1st. The steam pressure.

2d. The diameters of the cylinders.

3d. The stroke.

4th. The moving masses.

5th. The speed of revolution.

6th. The span from center to center of the after main bearings.

When inertia is taken into account, the worst case will be to suppose the L.P. engine with its heavier moving masses, acting on the after crank.

The minimum span from center to center of the after main bearings for a given stress in the material of the shaft, and definite nominal * pressures per square inch on the journals, is

* By nominal pressure is meant total reaction of bearing divided by product of diameter of journal into length of brass.

found by calculation from the first five factors given above; but as the disposition of these bearings is frequently affected by the exigencies of the design, it has been put down as a prime cause of stress in the shaft, and it has been deemed advisable to add a simple correction of the shaft dimensions given by the curves, for a change in this quantity from the minimum value.

The distribution of cylinder centers and the other main bearings, &c., by affecting the bending moments, has some influence on the maximum stresses in the shaft, but they are of quite secondary importance to the six factors noted above.

The whole investigation is practically founded on the following proposition with regard to similar machines. I will state it only with reference to similar steam engines, though that restriction is not necessary.

Two similar engines which have the same boiler pressure and the same distribution of steam pressures, all through, and which run at the same piston speed have:—

1st. The same intensity of stress in corresponding parts due to steam pressure.

2d. The same intensity of stress in corresponding parts due to inertia of the moving masses.

3d. Are similar in the same part of the revolutions: that is, have similar elastic deformations.

After this investigation was well under way the proofs of the 1st and 2d parts of this proposition were given in this JOURNAL for May, 1895, in the translation of a paper by Mr. J. A. Normand. The proof is retained here, as it is of fundamental importance in what follows.

(a) Suppose the linear dimensions of the smaller engine are to those of the larger, as . . . . . 1 : n.

(b) Total stresses due to steam at corresponding points of the revolution (proportional to piston areas) are as . . . . . 1 : n^2.

(c) Moving masses are as . . . . . 1 : n^3.

(d) Revolutions per minute (for the same piston speed) are as . . . . . n : 1.

(e) Inertia of moving masses at corresponding points of the revolution (proportional to product of moving masses, revolutions^2 and stroke = constant $\times$ $n^3 \times n^{-2} \times n$ = constant $\times$ n^2) are as . . . . . 1 : n^2.

(f) Arms for leverages for cross bending and torsion, are as . . . . . 1 : n.

(g) Areas for tension, compression, shear, and surface pressure on bearings, are as . . . . . 1 : n^2.

(h) Strength of the similar sections for cross bending and torsion, are as . . . . . 1 : n^3.

(i) Ratio of tensile, compressive, and shearing stress, and surface pressures on bearings per square inch, due to steam, ... = (b) $\div$ (g) = 1.

(k) Ratio of tensile, compressive, and shearing stress, and surface pressures on bearing per square inch, due to inertia, ... = (e) $\div$ (g) = 1.

(l) Ratio of stresses per square inch from cross bending and torsion, due to steam, ... = (b) $\times$ (f) $\div$ (h) = 1.

(m) Ratio of stresses per square inch from cross bending and torsion, due to inertia, ... = (e) $\times$ (f) $\div$ (h) = 1.

This proves the 1st and 2d parts of the proposition.

(n) Rate of extension, compression, and angle of shear at any part due to steam; that is, strain at any point (proportional to intensity of corresponding stress) . . . . . = (i) = 1.

(o) Rate of extension, compression, and

angle of shear at any part due to inertia; that is, strain at any point (proportional to intensity of corresponding stress) . . . . . $= (k) = 1.$

(p) Resistance of the similar sections to curvature due to cross bending forces, are as . . . . . $1 : n^4.$

(q) Curvature at corresponding points, due to cross bending from steam, That is, the *radius of curvature* is proportional to the linear dimensions, as it must be to give similar deformations. $= \frac{(b) \times (f)}{(p)} : 1 = n : 1.$

(r) Curvature at corresponding points due to cross bending from inertia. $= \frac{(e) \times (f)}{(p)} : 1 = n : 1.$

(s) Moment of resistance of similar sections, per unit rate of twist due to torsion couples, are as . . . . . $1 : n^4.$

(t) Twist per corresponding lengths from torsion, due to steam, . . $= \frac{(b) \times (f)}{(s)} : n = 1.$

(u) Twist per corresponding lengths from torsion, due to inertia, . . $= \frac{(e) \times (f)}{(s)} : n = 1.$

(q), (r), (t) and (u) may be combined in this single statement — At corresponding points in the revolutions, and corresponding points in the similar engines, the angular displacements of the parts are equal.

This completely proves the proposition.

The proposition does not hold if we consider forces due to gravity, but these are of slight importance in the consideration of the strength of crank shafts. A little consideration would show that, in the larger of the two similar engines, the stresses due to gravity are n times those in the corresponding parts of the smaller engine.

The use of this proposition allows us to confine our attention almost entirely to engines of one stroke.

In all that follows, unless the contrary is expressly stated, the

stroke is taken as being one inch. The symbols designating any quantity where the stroke ($=s$) is not one inch, are written with a prime, while in designating the corresponding quantity for a similar engine of one inch stroke the prime is omitted; thus,

$d'_1 =$ diameter of H. P. cylinder for stroke s.

$d_1 = \frac{d'_1}{s} =$ diameter of H. P. cylinder for stroke $= 1''$.

If the distribution of power between the three cylinders is nearly equal, and the cut offs in the several cylinders are not very different, we will have nearly equal maximum forces due to steam on each piston rod. It is almost invariably the intention to produce this approximate equality, and it is easily accomplished in practice over a considerable range of cylinder ratio for a given boiler pressure. If this maximum force were exactly the same on each rod, and if there were no losses of pressure from the boiler to the condenser (supposed at absolutely zero pressure) except due to expansion in the cylinders, its value can at once be calculated.

Let the cylinder diameters be d'_1, d'_2, ... d'_n, measured in inches, and taken in order beginning at the H.P., the steam passing successively through the cylinders.

Let $s =$ the stroke.

Let p, p_1, p_2, ... p_{n-1}, be the absolute boiler, and successive receiver pressures in pounds per square inch.

Then

$$\frac{\pi}{4}(p - p_1)\, d'^2_1 = \frac{\pi}{4}(p_1 - p_2)\, d'^2_2 = \ldots = \frac{\pi}{4} p_{n-1}\, d'^2_n = F'_m,$$

where F'_m is the maximum total force from steam on the piston rod.

Thus

$$p - p_1 = \frac{4}{\pi} \cdot \frac{F'_m}{d'^2_1},$$

$$p_1 - p_2 = \frac{4}{\pi} \cdot \frac{F'_m}{d'^2_2},$$

. . . . . .

. . . . . .

$$p_{n-1} = \frac{4}{\pi} \cdot \frac{F'_m}{d'^2_n}.$$

Adding,

$$p = \frac{4 F'_m}{\pi}\left(\frac{1}{d'^2_1} + \frac{1}{d'^2_2} + \ldots + \frac{1}{d'^2_n}\right),$$

or

$$F'_m = \frac{\pi p}{4\left(\frac{1}{d'^2_1} + \frac{1}{d'^2_2} + \ldots + \frac{1}{d'^2_n}\right)}.$$

The corresponding force for one inch stoke

$$= \frac{F'_m}{s^2} = \frac{\pi p}{4 s^2\left(\frac{1}{d'^2_1} + \frac{1}{d'^2_2} + \ldots + \frac{1}{d'^2_n}\right)}$$

$$= \frac{\pi p}{4\left(\frac{1}{d^2_1} + \frac{1}{d^2_2} + \ldots + \frac{1}{d^2_n}\right)} = F_m.$$

For three cylinders we have

$$F_m = \frac{\pi p}{4 s^2\left(\frac{1}{d'^2_1} + \frac{1}{d'^2_2} + \frac{1}{d'^2_3}\right)} = \frac{\pi p}{4\left(\frac{1}{d^2_1} + \frac{1}{d^2_2} + \frac{1}{d^2_3}\right)}. \quad \ldots \quad (1)$$

For the sake of reference in what follows, it is convenient here to explain Fig. VIII.

For the present, let us confine our attention only to the full line curves.

F_m forms the horizontal scale of this figure, and each curve gives by its ordinates read on the vertical scale, the dimension of the part of the crank shaft noted at the end of the curve, the stroke being one inch. For stroke s, each dimension must be multiplied by s. The curves are calculated from $F_m = 10$ to $F_m = 100$, which fully covers the range of practice. The lower values of F_m relate to engines in which the ratio of cylinder diameter to stroke is unusually small, or the steam pressure very low; while the large values of F_m relate to the opposite set of conditions.

ON THE COMPOSITION OF A TENSION AND SHEAR AT A POINT.

This is a subject of such great importance to what follows that it deserves to be noticed at length, even at the risk of repeating much that is well known—especially as a deviation from the ordi-

nary procedure in such a case, has been made in evaluating the factor of safety.

Suppose a cylindrical bar fixed at one end and supported horizontally. If a torsional couple is applied at the free end, each line in the bar which was originally straight and parallel to the axis will become a helix. In the outer fibers, where the greatest stress occurs, there will thus be a simple and uniform shear of the tangent plane at each point.

If, instead of a torsional couple, the free end of the bar is pulled vertically upwards, these longitudinal fibers will be extended with a force which is proportional to the distance of each point below the center of the cylinder, and also to its distance from the free end. Thus, at the lowest point of the fixed end of the bar, the longitudinal tension is a maximum.

If the torsional couple and vertical pull act together, the bar has at this point to stand the effect of the maximum shear and the maximum tension. At the upper point of the fixed end, this tension is replaced by an equal compression while the shear remains the same. If the bar is sufficiently long we may neglect all other stresses and these are the two points of highest stress.

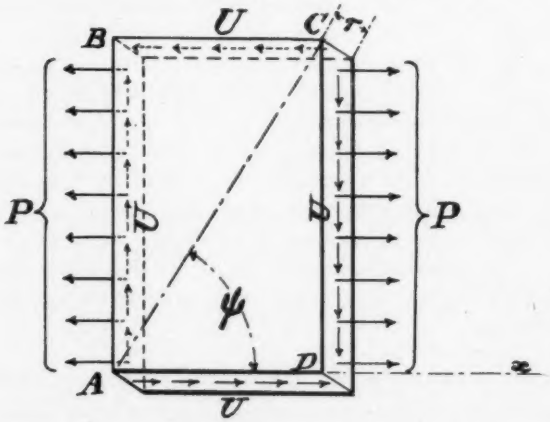


Fig. I.

Let us examine what occurs.

Suppose $ABCD$, Fig. I, to be a very small portion of the surface of the bar at the point where the longitudinal tension is a maximum.

This tension, P lbs. per square inch, acts parallel to Ax (which is the direction of the axis of the bar) on the two faces AB and CD. These two forces are thus in equlibrium. The shear of U pounds per square inch acts along all faces parallel to AB and CD. As the force is indicated in the figure, the portions of matter on the right of any face (say CD) tend to force the matter on the left of the face, in the direction BA. Thus the shearing forces acting on AB and CD produce a couple with moment $= U \times \tau \times AB \times BC$; τ being the very small thickness of the element.

For equilibrium, there must be an equal and opposite couple produced by shearing forces on the faces AD and BC.

As AB is the arm of this couple and $BC \times \tau$ is the area over which the force acts, the intensity of the shearing stress must also be U pounds per square inch.

Let us now examine the tension and shearing force across the diagonal AC. More properly, we should say, across the normal plane intersecting the element $ABCD$ of the surface in AC.

Let the angle $CAD = \psi$.

Considering the forces acting on the element ACD, we have:—

A tension $P \times CD \times \tau = \tau \times AC \times P \sin \psi$ parallel to Ax.

A shear $U \times AD \times \tau = \tau \times AC \times U \cos \psi$ parallel to Ax.

A shear $U \times CD \times \tau = \tau \times AC \times U \sin \psi$ perpendicular to Ax.

These must be balanced by the tension and shear across AC. Resolving the forces perpendicular and parallel to AC we get:—

Tension across AC,

$$\tau \times AC \times P_1 = \tau \times AC\,(P \sin^2 \psi + 2\ U \cos \psi \sin \psi).$$

Shear across AC,

$$\tau \times AC \times U_1 = \tau \times AC\,(-P \sin \psi \cos \psi - U \cos^2 \psi + U \sin^2 \psi).$$

Thus $P_1 = P \sin^2 \psi + 2\ U \sin \psi \cos \psi = \frac{1}{2} P\,(1 - \cos 2\psi) + U \sin 2\psi$. . . . . (2)

$$-U_1 = \tfrac{1}{2} P \sin 2\psi + U \cos 2\psi. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

When the tension (or compression) is a maximum we have,

$$\frac{d P_1}{d \psi} = P \sin 2\psi + 2U \cos 2\psi = O.$$

or,
$$\tan 2\psi = -\frac{2U}{P}. \quad \ldots \ldots \ldots \ldots \quad (4)$$

That is, the maximum tension and compression act along lines at right angles to one another.

Thus if $U = O$, equation (4) gives $\tan 2\psi = O$, from which $\psi = O$ or $\frac{\pi}{2}$: the latter value giving, as it obviously ought, P_1 $(= P)$ a maximum across the line AB.

If $P = O$, $\tan 2\psi$ is infinite, or $\psi = \pm \frac{\pi}{4}$.

The upper sign gives $P_1 = U$ and the lower $P_1 = -U$. Thus, as is well known, a shearing stress such as that figured, acting at the point A, is equivalent to a tension and compression of the same intensity per square inch acting across the two lines inclined at 45° to Ax.

If, then, P and U are both finite, the maximum tension will occur when ψ is between $\frac{\pi}{4}$ and $\frac{\pi}{2}$; that is, $\sin 2\psi$ will be positive and $\cos 2\psi$ negative. Therefore as

$$\text{Sin } 2\psi = \mp \frac{\tan 2\psi}{\sqrt{(1 + \tan^2 2\psi)}} = \pm \frac{2U}{\sqrt{(P^2 + 4U^2)}},$$

$$\text{Cos } 2\psi = \mp \frac{1}{\sqrt{(1 + \tan^2 2\psi)}} = \mp \frac{P}{\sqrt{(P^2 + 4U^2)}},$$

when those values are substituted in equation (2) the upper sign will refer to tension.

Using these values of $\sin 2\psi$ and $\cos 2\psi$ in equation (2) we get

$$P_1 = \frac{P}{2}\left(1 \pm \frac{P}{\sqrt{(P^2 + 4U^2)}}\right) \pm \frac{2U^2}{\sqrt{(P^2 + 4U^2)}}$$

$$= \frac{1}{2}[P \pm \sqrt{(P^2 + 4U^2)}].$$

Equation (4) reduces equation (3) to zero.

We have, then, as an equivalent stress system to that figured, a maximum tension

$$\frac{\sqrt{(P^2 + 4\,U^2)} + P}{2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

and a maximum compression

$$\frac{\sqrt{(P^2 + 4\,U^2)} - P}{2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

acting across two normal planes through A, which are at right angles to one another and inclined to Ax at angles given by equation (4)—and no shearing stress across these planes. This tension and compression are called the principal stresses at the point A; and the normal planes across which they act, the principal planes of stress.

If B is the bending moment producing the tension P, and T the torsional moment producing the shearing stress U, a being the diameter of the bar, we have within the elastic limit,

$$B = \frac{\pi}{32} P a^3 \text{ and } T = \frac{\pi}{16} U a^3.$$

The principal tension may therefore be expressed by

$$\frac{16}{\pi a^3}[\sqrt{(B^2 + T^2)} + B];$$

or the principal *tension* is of the same intensity as the *shearing stress* which would be produced by a torsional moment

$$T_e = \sqrt{(B^2 + T^2)} + B. \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

This is the usual formula for combining torsion and cross bending, and T_e is commonly called the "equivalent turning moment." But calculating the allowable stress on a shaft in this way entirely ignores the presence of the principal compression, which has a serious influence on the strength of the bar.

For, repeating somewhat, if $T = O$, there is no shearing stress and consequently no principal compression at the lower side of the bar, while

$$T_e = 2\,B = \frac{\pi}{16} P a^3;$$

and if $B = O$, $T_e = T$ and $P = O$, and the principal tension and

compression are each equal to the shearing stress U. In the first case we have,

$$T = O. \quad \text{Principal tension} = P = \frac{16\,T_e}{\pi\,a^3}.$$

$$\text{Principal compression} = O.$$

In the second case,

$B = O$. Principal tension $= U = \frac{16\,T_e}{\pi\,a^3}$, the same expression as for $T = O$ above.

$$\text{Principal compression} = U = \frac{16\,T_e}{\pi\,a^3}.$$

Now the amount of shearing stress which a square inch of wrought steel or wrought iron will stand at the elastic limit is only about seventy to eighty per cent. of the pull required to give it a sensible set; and, as the above shows, this lowering of strength is brought about by the compression applied at right angles to the direction of the line of tension and, no doubt partly also, if the material is æolotropic, by the direction of the principal planes of stress being changed.

Thus, for a given value of T_e, the factor of safety in the second, would only be from seventy to eighty per cent. of its value in the first case.

To determine the value of the ratio of shearing stress U at the elastic limit, to the tension P at the elastic limit, a considerable number of the published results of experiments on torsion, tension, and cross-bending have been examined. The results showed a range of the ratio U/P from considerably under 0.7 to over 0.8, though it seemed probable that a large part of the variation was due to imperfections in the experiments. Frequently this ratio has been taken as 0.8, but the experiments on the whole indicate that this value is decidedly too high. In this paper the ratio has been taken as 0.7, which at most seems to be a very few per cent. from the truth. It is improbable that the ratio is quite constant, especially for wrought iron, on account of its æolotropy. If the true value of this ratio is in any particular shaft higher than 0.7 it will have a greater factor of safety than is credited to it here.

It was intended to compile a table showing the values of the limiting elastic stresses and the elastic constants for various qualities of steel, but in most cases the data published was too meager; while, in others, the mode of experiment or values of some of the elastic constants deduced threw great doubt on the reliability of the experiments.

I have not found any experiments on the value of T_e at the elastic limit for different values of B/T, though the information they would afford would be of great practical value.

In what succeeds, instead of using equation (7), I have taken

$$T_a = \sqrt{\left[B^2 + \left(\frac{10\,T}{7}\right)^2\right]} + B, \quad . \quad . \quad . \quad . \quad (8)$$

calling this the "augmented equivalent turning moment" or, more shortly, the "augmented moment," to distinguish it from the usual method of combining torsion or cross bending. Unless the contrary is expressly stated, it is to be understood that the shearing stresses have been augmented in this way.

This expression for T_a does not in any way call in question the values of the principal stresses, and positions of the principal planes of stress given by equations (5), (6) and (4) respectively, but is merely an artifice for estimating more truly the value of the factor of safety for different values of the ratio B/T.

For $T = O$, or for $B = O$, the same value of T_a gives the same factor of safety. No doubt this is very approximately true, also, over a considerable range of the ratio B/T, when either B or T produce the large proportion of the stress. When B and T are near an equality it seems very probable that equation (8) will allow us to estimate the factor of safety much more truly than equation (7). The equation (8) can, however, only be looked on as tentative in the absence of the very simple experiments from which a practically true law for T_a could be easily deduced.

If we are combining the tension P and shearing stress U, the corresponding equation to (8) will, of course, be

$$U_a = \sqrt{\left[\left(\frac{P}{2}\right)^2 + \left(\frac{10\,U}{7}\right)^2\right]} + \frac{P}{2} \quad . \quad . \quad . \quad . \quad (9)$$

This may be called the "augmented stress" at the point.

Perhaps the simplest of the many ways of making the experiments on combination of torsion and cross bending would be to fix a cylindrical test piece in the machine and, on this as an axis, mount a grooved pulley round which a wire is wrapped for applying the test pull. By clamping the wheel to the bar at a properly chosen point, any value of the ratio of B/T can be readily obtained, which would remain constant till the bar was sensibly bent; this, if proper proportions are chosen, need not be before the elastic limit is reached. To obtain the considerable number of bars required for one series of experiments, they might, in the case of the malleable metals, be cut longitudinally from the circumference of a larger cylindrical forged bar. This would insure, as far as possible, if the forging is carefully conducted, the production of test pieces of the same material which had received the same treatment, and whose quantities were, therefore, comparable. The behavior of the outer portions of round bars is, also, the most important case to investigate.

Equation (4) shows that the angular positions of the principal planes of stress vary with each change in the ratio U/P; consequently, we would expect a very different law to result from experiments on the combination of cross bending and torsion, according as they were made on a very decidedly æolotropic substance or on one which was nearly isotropic.

STRESSES DUE TO STEAM ALONE.

It is convenient to consider first the stresses in the crank shaft which are caused by steam pressure alone. These are the stresses which would be produced if all the moving parts were massless, or if the engine ran so slowly at full power that the inertia of the moving parts were inappreciable. The stress produced at any point of the shaft varies during the revolution; and the different parts of the shaft, as for instance the aft main journal and the crank pin, come to their maximum stress at entirely different positions of the crank, as a rule—positions which depend largely on the distribution of steam in the cylinders. In the three-crank engines we are considering it will

become evident as we proceed that, at those points of the revolution at which steam alone would produce a maximum stress at any point of the shaft, the inertia of the parts would decrease this stress. Thus, this maximum is the limit beyond which the stress cannot rise in engines where the speed is kept below a limit which is definite in each particular case—a limit which will naturally appear in the second part of the investigation.

When this limit of speed is exceeded, the steam and inertia combine to produce a greater maximum stress than that due to steam alone, but this occurs at a different part of the revolution from the previous maximum. The consideration of this second maximum we leave for the present.

When below the limit of speed above referred to, it is safest to neglect entirely the alleviation of stress due to inertia, as the maximum limit of stress due to steam may always be somewhat nearly attained; especially if the engine is retarded under full steam, as may occur in any case.

In all that follows, unless special exception is made, we shall suppose the following:—

(a) That the engines are of the inverted vertical type.

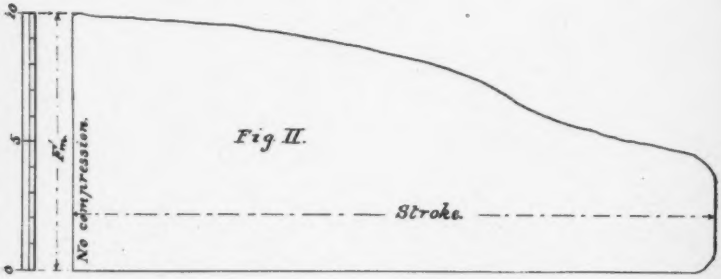
(b) That the cylinders are in the order H.P., I.P., L.P., the L.P. being aft.

(c) That the length of connecting rod, center to center, is four times the length of the crank radius.

(a) and (b) avoid circumlocution. (b) has no influence on the steam stresses, if the indicator diagrams for the three cylinders are exactly the same; but having the L.P. aft is the worst case where inertia is considered. (c) is the worst proportion frequently used, and in no case is the connecting rod much shorter than this.

Fig. II is the indicator card which has been taken as fairly representing what occurs in practice at full power. The compression has been omitted, as that is a very variable element which is almost always small at full power. If of only moderate amount its influence on the maximum stress, whether steam stress alone or steam and inertia stresses are being considered, is exceedingly trifling. Its effect on the turning moment from any

cylinder, is to diminish it just before the end of the stroke, where that turning moment is already inconsiderable. Its effect on the cross bending stresses near the aft end of the shaft is only sensible in the case of the L.P. cylinder, in which the compression at full power is usually almost invisible.



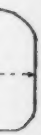
Standard Indicator Diagram.

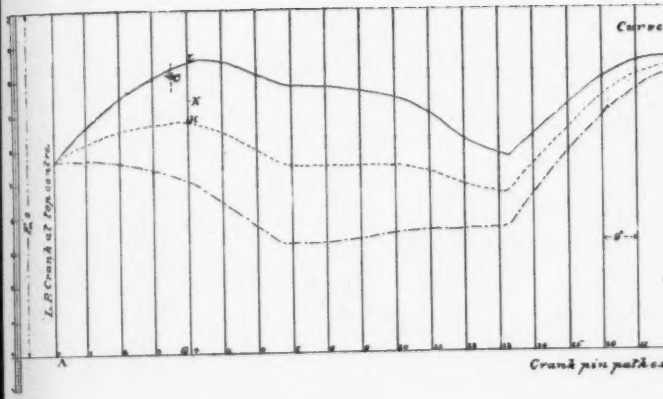
From this indicator card in each cylinder, the maximum total force on the three piston rods being the same, the turning moment curves in Fig. III have been deduced. The base *AB* is the crank pin path expanded. The ordinates of the full curve give the sum of the turning moments from the three cylinders. The dot-and-dash line curve is that from the H.P. and I.P. together, while the dotted line gives the sum of the H.P., I.P., and one-half of the L.P. turning moments.

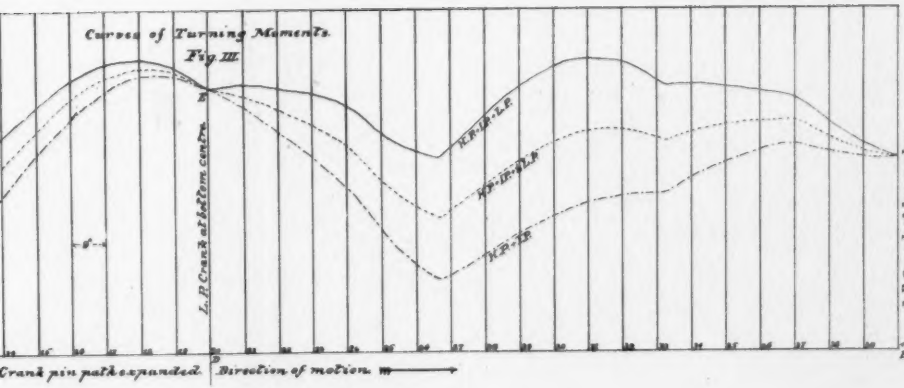
Ten divisions of the end scale represent the moment which would be produced by the maximum force on the piston rod multiplied by the stroke ($= F'_m s$).

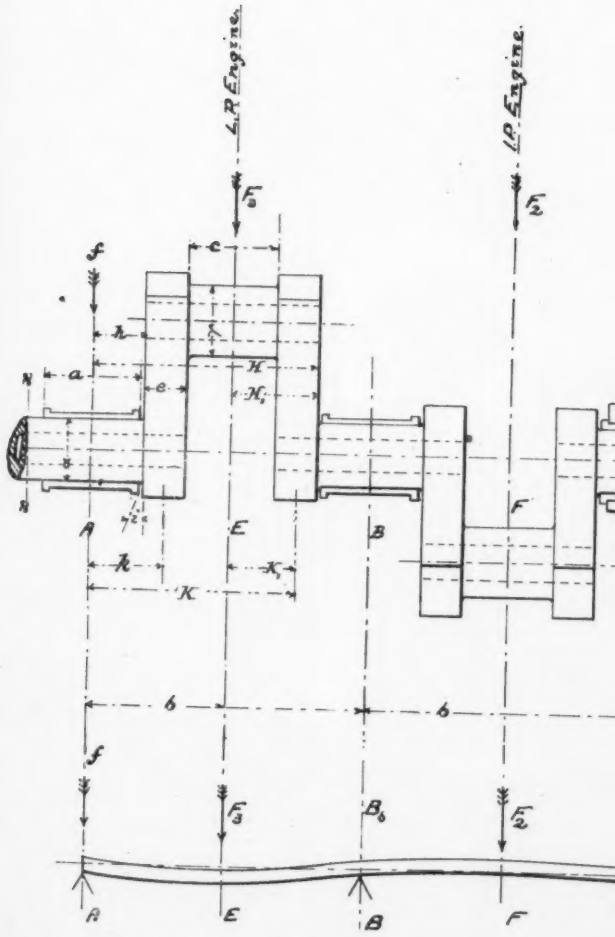
AB represents one revolution beginning and ending when the L.P. is at top stroke. The whole length *AB* is divided by ordinates into forty equal parts, each of which thus represents 9° of the crank circle.

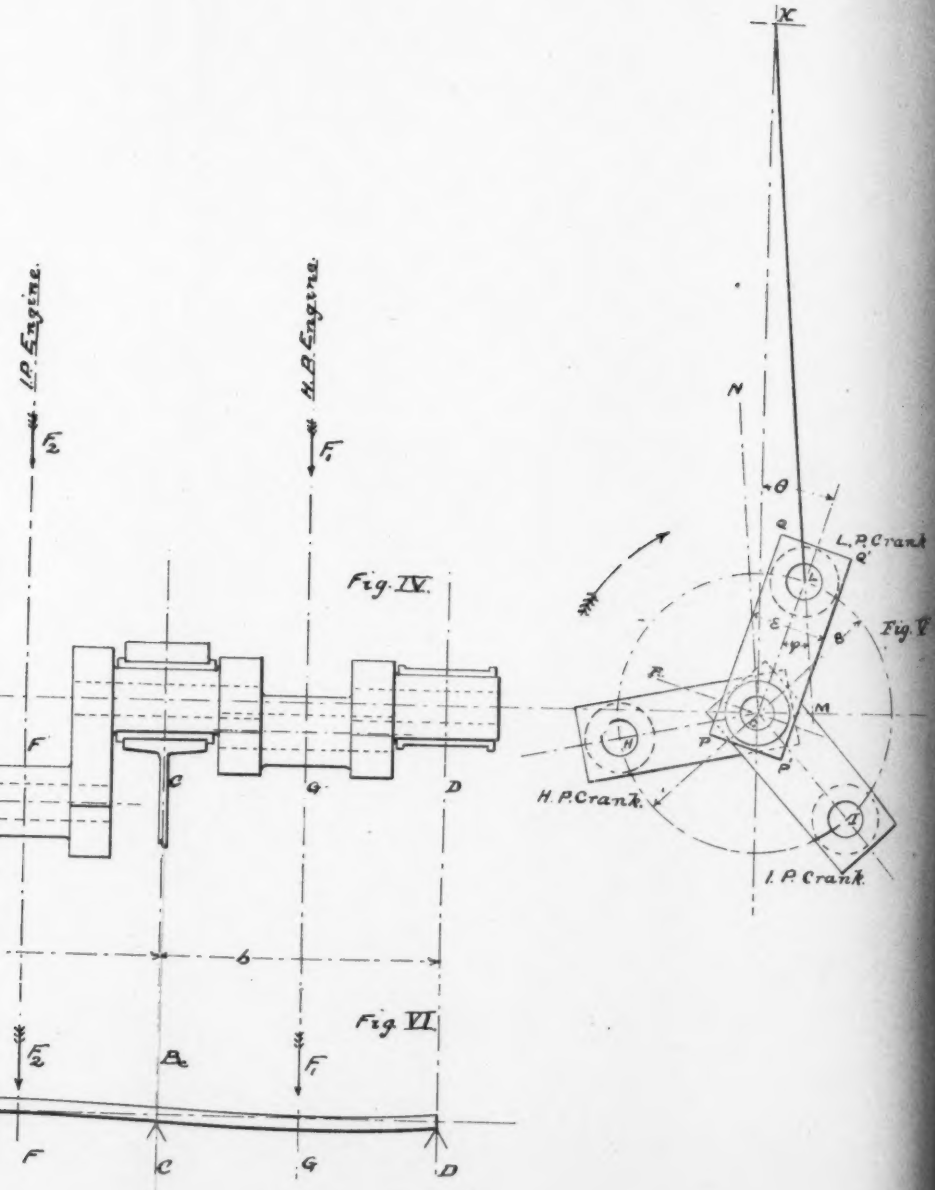
We have now to estimate the bending moments. This has been done on the supposition that no bending moment is transmitted through the section *xx*, Fig. IV, from the crank shaft to the line shafting. If there is a bearing in the line shafting near




L.P. Crank at top centre.
Crank pin path.


Curves of Turning Moments.
Fig. III.
H.P.&L.P.
H.P.&L.P.
H.P.
L.P. Crank at bottom centre.
L.P. Crank at top centre.
Crank pin path expanded.
Direction of motion.


L.P. Engine.
L.P. Engine.
F_3
F_2
f
c
h
z
H
H_1
N
a
e
s
z_1
z_2
A
E
B
F
k
k_1
K
b
b
f
F_3
B_b
F_2
A
E
B
F


L.P. Engine.
H.P. Engine.
F2
F1
Fig. IV.
F
C
G
D
b
Fig. VI.
A
F2
F1
F
C
G
D
X
N
θ
Q
Q'
L.P. Crank
Fig. V
ε
β
R
M
P
P'
H.P. Crank.
I.P. Crank.

the aft L.P. bearing A, and in line with the crank shaft bearings, the bending moments in the crank shaft will be less than for zero bending moment at xx; but a little want of alignment of this line shaft bearing would at once realize the case supposed. Again, the four bearings A, B, C, D, are supposed equally spaced center to center, and the cylinder centers E, F, G, equally spaced between the centers of the main bearings. The crank shaft is cut off forward of D.

It can be verified by calculation from the dimensions of the crank shafts which will be deduced that, when the bearings are in line, the parts where the heaviest stresses are produced, are: 1st. In the aft L.P. journal where it joins the aft L.P. web. 2d. In the aft L.P. web. 3d. In the L.P. crank pin. That being the case, it is for these parts that the dimensions of the shafts must be determined. These are the points of greatest stress even when the bearings are somewhat out of line. But, as the forward L.P. web and forward L.P. main journal are farther from the aft main bearing, the increased reaction on which produces the additional bending moments on the parts of the L.P. crank when the shaft is out of line, the stresses on these parts rise more quickly than on those further aft. In what immediately succeeds, no allowance is made for want of alignment, but the question will be carefully considered later.

The exact problem of the elasticity of a crank shaft, such as is shown in Fig. IV, is of insuperable difficulty. To express even approximately the influence of the webs introduces considerable complications. The general effect is, for most practical cases, to diminish the stiffness of *each* span between the main bearings, from what it would be for a straight bar in which the length occupied by the webs is replaced by a piece of the same diameter as the main journals. For those shafts in which the stroke is long compared with the span of the main bearings (F_m small) this diminution of stiffness is very marked, but this effect of the webs rapidly diminishes as the diameter of shaft and span of main bearings has to be increased through increased force on the piston rod. It is not more than a few per cent. in error to calculate the reaction on the aft main bearing from the simpler

case of an originally straight uniform bar resting on four points, as shown in Fig. VI.

No doubt this supposition gives the arm for the maximum bending moments a value slightly in excess of the truth.

In Fig. VI, let F_1, F_2, F_3, be the variable downward pressures at G, F and E; that is, the variable pressures on the H.P., I.P. and L.P. piston rods, the same letters having the same meanings in Figs. IV and VI. All forces are reckoned positive when they act downwards on the shaft, and bending moments are positive when the upper fibers of the shaft are in tension.

Take B_b and B_c as the bending moments induced by these forces at B and C, and let the bending moments at A and B be zero; also suppose bearings A, B, C and D, as in line.

Let f be the reaction of the aft bearing point A.

Now it is known that for F_1 acting alone we have*

$$\left.\begin{aligned} B_b + 4\,B_c &= \tfrac{3}{8}\,F_1\,b. \\ 4\,B_b + B_c &= O. \\ \therefore 15\,B_b &= -\tfrac{3}{8}\,F_1\,b, \text{ or } B_b = -0.025\,F_1\,b; \\ \text{and} \qquad 15\,B_c &= \tfrac{3}{2}\,F_1\,b, \text{ or } B_c = 0.1\,F_1\,b. \end{aligned}\right\} \quad . \quad (10)$$

If F_2 acts alone

$$\left.\begin{aligned} 4\,B_b + B_c &= \tfrac{3}{8}\,F_2\,b = B_b + 4\,B_c, \\ \text{and} \qquad B_b &= B_c = 0.075\,F_2\,b. \end{aligned}\right\} \quad . \quad . \quad . \quad . \quad . \quad (11)$$

Similarly, if F_3 acts alone, we get

$$\left.\begin{aligned} B_b &= 0.1\,F_3\,b, \\ B_c &= -0.025\,F_3\,b. \end{aligned}\right\} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

Thus, if all three forces act together, we get from equations (10), (11) and (12)

$$\left.\begin{aligned} B_b &= (-0.025\,F_1 + 0.075\,F_2 + 0.1\,F_3)\,b, \\ B_c &= (0.1\,F_1 + 0.075\,F_2 - 0.025\,F_3)\,b. \end{aligned}\right\} \quad (13)$$

To estimate the reaction f at A we have, taking the moments acting on the span $A\,B$ about the point B,

$$f\,b + F_3\frac{b}{2} = B_b = (-0.025\,F_1 + 0.075\,F_2 + 0.1\,F_3)\,b,$$

or

$$f = -0.025\,F_1 + 0.075\,F_2 - 0.4\,F_3. \quad . \quad . \quad (14)$$

* See "A Treatise on the Mathematical Theory of Elasticity," by A. E. H. Love, M. A., Cambridge: at the University Press, Vol. II, p. 39.

This is the equation from which the curves of reaction on the aft bearing, shown in Fig. VII, have been drawn.

As in Fig. III, $A\,B$ is the crank path expanded, the L.P. crank being at top center at A and B, and ordinates are drawn for every 9° of the crank circle.

The reaction at the aft bearing will depend on the order of cranks; that is, on the direction in which the engine is running for the crank shaft under consideration.

The value of f is thus laid off on both sides of AB, one set of curves referring to one order of cranks, and the other set to the other, as marked. As the L.P. engine is the principal factor in producing this reaction, the value of f is negative on the down stroke and positive on the up stroke, except at the very ends, for both orders of cranks.

The dotted line gives the reaction due to the L.P. engine alone, the dot-and-dash line that due to the L.P. and I.P., and the full line that due to all three engines.

Ten divisions of the end scale give the value of F'_m.

After the whole work was practically completed, a new investigation of the distribution of pressure on the bearings was made which took almost exact account of the influence of the webs. When applied to the shafts given by Fig. VIII, it was found that the differences it would have made in Fig. VII were entirely unimportant. Obviously the maximum differences occur when $F_m = 10$. In this case the reaction f, when a maximum, was reduced about $2\frac{1}{2}$ per cent. The simpler investigation was therefore retained.

The torsional yield of the crank pins and the bending of the webs in a transverse plane, under the action of the turning moments induced in these parts, increases the elasticity of the shaft. The effect on f is to reduce slightly the influence of F_1 and F_2, and increase that of F_3 in equation (14), the changes depending somewhat on the point of the revolution considered. It will be found as we proceed that the stress in the shaft is usually a maximum when the L.P. crank is at or near one of the dead centers, and at these positions the total change of f produced by the above action of the turning moments is negligible.

Any other distribution of bearings and cylinder centers would not have led to a sensibly better result than that shown in Fig. VII. At first sight it would appear that putting two bearings between each crank, as so often has to be done, would reduce the bending moments on the L.P. span; but here again the slightest want of alignment, even the running clearance, would reduce the two cases to an equality. The position of the H.P. engine and bearing D have only a very small influence on f.

When F_2 is positive and F_3 is negative they both produce a positive value of f, and vice versa; and at those points of the revolution for which a maximum stress occurs, F_2 and F_3 are of opposite sign. As shifting forward the center F and bearing C will obviously diminish the influence of F_2, and increase that of F_3, on f, the total change of f will thus be the *difference* of these two changes and will be small.

We may then, with ample accuracy, accept the reactions shown in Fig. VII for any ordinary distribution of cylinder centers and bearings.

The horizontal forces from the connecting rod have not been taken into account in equation (14). In a single crank engine the greatest increase of bending moment, if the horizontal forces were taken account of, would of course be in the proportion of the total force on the connecting rod to the total force on the piston rod: that is (for a connecting rod which is four cranks in length) a maximum increase of about 3 per cent., which would not occur exactly at the points of the revolution for which the stresses in the shaft are a maximum. In the case we are considering the effect on the bending moments would be much less even than this, and is therefore neglected.

The forces and moments induced in the aft main journal, aft web, and L.P. crank pin are:—

AFT MAIN JOURNAL.

Turning moment.—This is given by the full line in Fig. III. Friction in the journals forward, which will diminish this turning moment slightly, is neglected.

Bending moment.—This is greatest where the journal meets the web. It is taken as $f \times h$, h being the distance from the aft side of the web to the center of the journal A.

Figs. III and VII at once show that the bending and turning moments will combine to produce a maximum stress in the main journal when the L.P. crank has passed top center by 36° to 40°. The bending moment at this point is somewhat greater for the order of cranks H.P., L.P., I.P., than for the other. As a reversal of the engine changes to the other order of cranks in the three-crank engine, the moments should always be measured for that order which gives the worst stress, even if one order is taken at one point and the other at another in calculating the same shaft.

Shearing stress.—There is also a total shearing stress distributed in an ununiform manner over the section of the shaft. This shearing stress has a large effect in shifting circumferentially the point of maximum combined stress in the shafts about to be calculated, but its effect in raising that maximum is not in any case more than $3\frac{1}{2}$ per cent. It is consequently neglected in calculating the main journal diameters, though a uniform increase of one per cent. in these diameters would almost exactly make that allowance. As the distribution of shearing stress over the section of a circular shaft, solid or hollow, is of considerable importance in practical mechanics quite apart from its bearing on the present subject, the curves drawn for this investigation are given toward the end of the paper.

AFT CRANK WEB.

Turning moment.—The turning moment in the aft journal is transmitted from the aft web, and appears in the web as a transverse moment producing a cross bending strain in a transverse plane. The maximum tension and compression thus arising occur in the longitudinal surfaces of the web shown in projection in FQ and $F'Q'$, Fig. V. The intensity of stress in PQ and $P'Q'$ will increase as we pass from the crank pin to the journal, until a point is reached where the presence of the journal relieves the web from further increase. This is probably not far from the plane OR (at right angles to OL) as PQ and $P'Q'$ lie altogether out-

side the journal. In all calculations below, this point is taken as distant from OR by about one tenth of the crank radius. Consequently the turning moment is equal to that on the main journal less one-tenth of the difference of turning moment on the main journal and crank pin. This no doubt slightly overestimates the moment for those crank shafts in which the main journal diameter is a large proportion of the crank radius, but the error is altogether unimportant.

The moment given in Table I column III, line (d), is measured at C, Fig. III.

Bending moment.—The reaction f from the aft bearing A produces a bending moment. If the direction of f were ON, Fig. V, parallel to the connecting rod KL, the value of this moment would be $fk \cos \varphi$, where φ is the angle OLM between the connecting rod and crank, and k is the distance from the center of the bearing A to the center of the web. This moment will produce a maximum tension and compression in the forward and aft sides of the web.

These two moments, as we shall see, are the only two factors in producing stress which we need take account of. As the first produces its maximum stress at the surfaces PQ and $P'Q'$, Fig. V, and the second on the forward and aft sides of the web, the maximum combined stress occurs along two diagonally opposite *edges* of the webs. The stress comes to a maximum for the web a trifle earlier in the revolution than for the main journal; as, when the L.P. crank is about 36° past top stroke, the bending moment in the web is quickly diminishing on account of the increasing angle (diminishing value of cos φ) between the connecting rod and crank.

Examination of Fig. V will show that, because of friction and the reaction from the horizontal force of the intermediate connecting rod on bearing A, the line of reaction ON will be turned round somewhat from the position shown, toward OL; also, the want of equality in the distribution of power between the engines, may bring the maximum combined stress earlier in the revolution than is given by Figs. III and VII. Therefore the bending moment $fk \cos \varphi$ when the total stress is a maximum,

TABLE I.—MOMENTS STRESSING SHAFTS.

Line.	Column.	I.	II.	III.	IV.	V.	VI.
(a)	Part of crank shaft.....	Aft main journal.		Aft crank web.		Aft crank pin.	
(b)	Moment.....	Turning.	Bending.	Turning.	Bending.	Turning	Bending.
(c)	Point of revolution at which moment is measured	L.P.	crank about	36° past top center.		L.P.	crank at bottom.
(d)	Value of moment from Figs. III and VII.....	$.85\ F_m$	$.459\ F_m\ h$	$.868\ F_m$	$.9 \times .464\ F_m\ k$ $=.418\ F_m\ k$	$.77\ F_m$	$.481\ F_m \times b/2$ $=.2405\ F_m\ b$
(e)	Ten per cent. loss from boiler pressure to absolute zero	.9	.9	.9	.9	.9	.9
(f)	Increase for unequal steam stresses.....	1.06	1.1	1.06	1.1	.....	1.1
(g)	Increase for errors of adjustment.....	.....	1.1	.....	1.1	.....	1.05
(h)	Shearing stress augmented in ratio 7 to 10...	10/7	.....	.....	.....	10/7	.....
(k)	Increase for change of section.....	1.05	1.1	1.1	1.1	.....	.....
(l)	Value of moment taken in calculation.....	$1.216\ F_m$	$.55\ F_m\ h$	$.85\ F_m$	$.50\ F_m\ k$	F_m	$.25\ F_m\ b$

Multipliers.

The moments in Table I are for a stroke of one inch. The turning moments for stroke s will be got by changing F_m into $F'_m\ s$ in columns I, III, and V, and the bending moments by changing $F_m\ h$, $F_m\ k$, and $F_m\ b$ into $F'_m\ h'$, $F'_m\ k'$, and $F'_m\ b'$ in columns II, IV, and VI, respectively. The bearings are supposed to be in line.

instead of being about 0.71 fk, which is its value when $\theta = 36°$, has been taken as 0.9 fk to cover the possible change in the angle between the line of reaction f and center of crank web at time of maximum stress.

The other stresses in the aft web are:—

1st. A tension or compression arising from the reaction f, distributed over the section of the web. This never raises the maximum stress, allowed below, more than 3½ per cent., and is neglected.

2d. A total shearing stress equal to $f \sin \varphi$ (supposing ON the direction of the reaction f) ununiformly distributed over each cross section of the web. As will be evident when we have considered the distribution of shearing stress over a circular section, it is zero at the surfaces PQ and $P'Q'$, Fig. V, and consequently at the points of maximum combined stress for the turning and bending moments. As it is of low average intensity it will not have any effect on the position or intensity of that maximum.

3d. A torsional moment $fk \sin \varphi$ producing shearing stress. This stress is zero along the longitudinal edges of the webs, and is a maximum at the points on the surface midway between these edges. That is, Fig. IX being a cross section of the web, the shearing stress is zero at the angular points A, B, C, D. It is a maximum at E and F, the middle points of the long sides. There is a smaller maximum at G and H, the middle points of the short sides. It is remarkable that the combination of this shearing stress with the stresses due to turning and bending still leaves, in all cases calculated below, the maximum augmented stress at the angular points of the web. Consequently, this torsional moment has no influence whatever on the strength of the shafts. In the worst case for steam alone ($F_m = 100$) when the shaft is in line, the maximum augmented shearing stress at E and F rises to about 23 per cent. of the maximum augmented stress allowed in the material.

Various curves relating to the torsion of a rectangular prism, and the necessary explanations, are retained at the end of the paper, as the subject is of great importance.

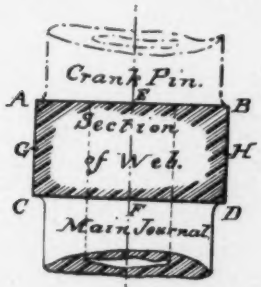


Fig IX.

CRANK PIN.

Turning moment.—The turning moment transmitted across any section of the shaft is here considered positive when the tendency is to accelerate the motion of the part of the shaft aft of the section, in the direction of revolution of the engine. Of course the moment acts oppositely on the part of the shaft forward of the section—when positive, tends to retard it.

Now consider the forces acting on the shaft between the section xx, Fig. IV, and a section at E—the center of the L.P. engine. Take moments about L, Fig. VI, the center of the L.P. crank pin.

1st. There is a positive moment, given by the full line Fig. III, transmitted across xx.

2d. The reaction f, if acting along ON, Fig. V, parallel to KL, will produce a moment at L equal to $f \times OL \sin \varphi$. As Fig. V is drawn, f acts upwards on the shaft, and this moment tends to accelerate the parts forward of E. It is, therefore, negative during the L.P. down stroke. Similarly it is negative on the upstroke.

3d. The action of the connecting rod on the pin aft of E produces no moment about L, except an unimportant frictional one.

4th. The only other moment is the one we wish to measure, acting at L; and it is thus seen to be equal to the whole moment from the three engines, diminished by $f \times OL \sin \varphi$. As f, when large, does not differ much from $F_3/2$, and as the L.P. turning moment is very nearly $F_3 \times OL \sin \varphi$, the total moment on the crank pin is approximately the sum of the H.P. and I.P. turning moments together with half the L.P. turning moment. The dotted line, Fig. III, was thus drawn to indicate roughly where the maximum stress would come.

Bending moment.—If the pressure on the crank pin and main journal were concentrated at E and A, the value of this bending moment would be $f \times b/2$. But the distribution of pressure on the crank pin reduces this from 10 to 12 per cent. when everything is in proper adjustment. There is an allowance explained below which again increases this moment.

At the center of the pin there is no shear of any importance.

An inspection of Figs. III and VII shows that the maximum stress in the crank pin occurs when the L.P. crank is at bottom center. Just before that point the turning moment is a little larger, but the bending moment is much smaller. The reaction f produces no sensible effect on the turning moment; especially as, at this part of the revolution, friction and the horizontal forces from the I.P. connecting rod affect ON, the line of reaction of f, oppositely.

In Table I is given the deduction of the values of the various moments showing all the allowances which have been made.

Line (d) gives the values of the moments from Figs. III and VII, taken at the points indicated above. In column VI the value is that of $f\,b/2$.

Line (e). In every engine there is a fall of pressure between the boiler and H.P. initial pressure. A further loss takes place between each cylinder, and between the L.P. cylinder and condenser, the latter having always a finite pressure. All these losses of pressure go to reduce the moments in line (d). The total loss is taken as 10 per cent. of the absolute boiler pressure, which is less than almost ever occurs at full power.

Line (f). The supposition made at the beginning of the paper was, that there was a fairly equal distribution of power among the three cranks. This condition may be held to cover the case of any one of the engines giving 10 per cent. more power than it would do when the distribution is exactly equal. If this increase of power is in the L.P., all bending moments would be increased about 10 per cent.

It has also been supposed in Table I that the increase of power in the L.P. is altogether at the expense of one of the other cylinders—the H.P. for order of cranks H.P., L.P., I.P. Thus, at the point of maximum stress for the aft journal and crank web, the sum of the turning moments from the H.P. and I.P. is not sensibly decreased, since that from the former is then very small; and 10 per cent. increase of the L.P. turning moment will raise the ordinates of the full line curve, Fig. III, from 4 to 6 per cent. near ordinate 4. Thus 1.06 is the multiplier in columns I and

III. Indeed this figure would hardly be reached without some increase of power in the I.P. also.

The case supposed would increase the bending moments in the crank pin at bottom center 10 per cent., but would slightly decrease the turning moments. This decrease has been neglected and no multiplier appears on this line in column V. If the increase of power had taken place in one of the other cylinders, but not at the expense of the L.P., the case would have been a trifle worse than that supposed for the L.P. crank pin. The greatest increase over the stresses due to the moments in Table I would in this case occur when $F_m = 10$, and even here it would only have been about 4 per cent.

Some cases of very bad distribution of steam will be discussed later, when it will be found that in no case at all likely to occur, is the stress much increased above that due to the moments in Table I.

Line (g). It was impossible to anticipate what increase of bending moment would have to be allowed in any particular case for a given want of alignment, until the dimensions of the shaft were known. Judging from Mr. Milton's excellent papers on crank shafts,* the allowance required is large, but the difficulty of making it is increased by the fact that in two similar engines (that is, for a given value of F_m) the proportionate increase of bending moment required is greater in the smaller than in the larger engine. For if we consider two similar engines, one of 10 and one of 20 inches stroke, the proposition with regard to similar engines at once shows that a want of alignment in the aft main bearing which in the small engine amounts to $\frac{1}{100}$ of an inch, produces the same increase of stress in, say, the crank pin, as an error of alignment of $\frac{1}{50}$ of an inch in the larger engine. Accuracy of workmanship does not follow the same law. Indeed, a good workman would align the 20, almost as truly as the 10 inch stroke engine. Again, every shaft may be considered out of line to the extent of its running clearance, but this quantity in the larger engine would by no means be twice that in the smaller. The most satisfactory course, then,

* See *Transactions* of the Institution of Naval Architects, Vols. XX and XXII.

seems to be to calculate the dimensions of the shaft for the bearings in line—afterwards investigate what change of stress is produced by a given want of alignment—then deduce the change of shaft dimensions, if any are required.

Imperfect bedding of the shaft, or undue cutting of the brass from heating or other causes, may shift the center of pressure fore and aft and thus increase the arm of the bending moment. The factors in line (g) are introduced to cover this slight effect. The increase in columns II and IV is 10 per cent., and, in column VI, actually about 15 per cent.; for it will be remembered, the moment in line (d) column VI is overestimated fully 10 per cent. The greater value is given in this case, as the distribution of pressure on the crank pin brass may add to the increase of moment.

Line (h). This increase of the shearing stress is fully explained above.

Line (k). Where the round parts of the shaft meet the webs, the presence of the large and consequently rigid mass of metal has some influence on the distribution of stress. Its exact influence would be extremely difficult even to approximate to by calculation, and experiments on the subject seem hardly to exist. If there is no fillet at the junction, the maximum stress due to cross bending or a simple longitudinal pull of the round part of the shaft is very seriously raised, while the presence of a moderate fillet largely prevents this rise. Every one knows how essential is the fillet under the head of a bolt. Some experiments by Mr. David Kirkaldy,* show that, in a round bar one part of which is one square inch and another two square inches in section, the presence of a small fillet may obviate local rise of stress altogether when the bar is pulled asunder; though no certain conclusion could be drawn from the small number of experiments published. The result might be somewhat different where a round shaft joins a large slab of metal such as the crank web.

* See "Strength and Properties of Materials," by W. G. Kirkaldy, London; Samson Low, Marston, Searle and Rivington, Ltd. Also see "Engineering," Vol. XI, page 261, for experiments by Wöhler on similar bars.

The allowance made in column II is 10 per cent. That this increase of maximum stress ascribes very considerable influence to change of section, will be obvious if we remember that the total moment in the section cannot be affected—being equal to the force on the main journal acting at a definite leverage. Therefore, the rise of maximum stress, which occurs in the outer fibers farthest from the neutral axis of the section, involves a fall of stress in the parts nearer the neutral axis.

For torsion the stresses would also be raised, as the outlying masses of metal in the web would tie more rigidly the outer than the inner portions of the round part at the junction. The influence would, no doubt, be much less than in the case of cross bending and, consequently, a smaller multiplier has been inserted in column I. The shear due to torsion in the webs, mentioned above, has only an unimportant influence on the shear in the round parts of the shaft.

It is more difficult to tell what is the influence of the change of section on the stresses in the web. Consequently, to meet any possible rise at the edges where the maximum stress occurs, the multipliers in Columns III and IV were inserted. I think, however, that we may safely conclude that, for the cross bending stresses, the presence of the main journal would somewhat increase the stress near F, Fig. IX; and consequently rather reduce it at C and D—since again the total moment is not affected—while its influence on the side AB would be inappreciable. Similarly, the stresses at C and D, due to turning moment, are probably lowered slightly, and the stresses at A and B unaffected. We may conclude then, that the stress in the webs is overestimated by 10 per cent.

The maximum stress in the crank pin occurs at the center, usually. If the influence of change of section is that assumed, the stress at the aft end of the pin is very slightly higher than at the center for the lower values of F_m, but the difference is practically nothing. Thus in columns V and VI there are no multipliers in line (k).

By using the multipliers in lines (e) to (k), the moments given in line (d) are changed into those in line (l).

Each of these multipliers makes an allowance for a factor which is always present, and affects the strength and life of the shaft. They affect the moments so that the ratios of those in line (d) are different from that of those in line (l) (if it were not so they would have no influence on the final result); and the determination of constants and subsequent calculation of shafts given in Fig. VIII, by means of the values in line (l), is no doubt much better than by using the values in line (d).

The total pressures on the crank pin and aft main journal, with multipliers similar to those in Table I, are given in Table II. The measurements from Fig. VII are for the order of cranks H.P., I.P. and L.P.

TABLE II.—TOTAL PRESSURES ON CRANK PIN AND AFT MAIN JOURNAL.

Line.	Column.	I.	II.
(a)	Part of shaft	Crank pin.	Aft main bearing.
(b)	Point of revolution at which pressure is measured.	L. P. crank at top or bottom center.	
(c)	Value of total pressure from Figs. II and VII.	F_m	$.482\ F_m$
(d)	Multipliers. Ten per cent. loss from boiler pressure to absolute zero	.9	.9
(e)	Multipliers. Increase for unequal steam stresses	1.1	1.1
(f)	Multipliers. Increase for errors of adjustment	.....	1.1
(g)	Value of total pressure taken in calculation	$.99\ F_m$	$.525\ F_m$

The pressures in Table II are for a stroke of one inch. The total pressures for stroke s will be got by changing F_m into F'_m—that is, multiplying by s^2.

Table III gives the data for deducing the constants to be used in the calculations. The ten engines used had all successfully passed their trials and many had been a long time in commission at the time the investigation was begun. They are all triple expansion engines with three cranks at 120°.

In column VI the lengths given are those of the aft L.P. main bearing, except in line (a) where the lengths of both the forward H.P. and aft L.P. main bearings are given, as noted.

In column XIII, in those cases where the forward and aft L P. main bearings are of different lengths, the dimension given as the span of the main bearings is the sum of the clear distance

TABLE III.—DATA FROM ENGINES OF U. S. NAVY.

Line.	Column.	I.	II.	III.	IV.	V.	VI.	VII.	VIII.	IX.	X.	XI	XII	XIII.	XIV.
	Ship.	Boiler pressure above atmosphere.	Cylinder diameters.	Stroke of engine.	Main journal.			Crank pin.			Crank webs.		Arm for bending moment on main journal $= k'$.	Span of main bearings $= b'$.	Arm for bending moment on webs $= k'$.
					Outside diameter.	Inside diameter.	Length of shortest bearing.	Outside diameter.	Inside diameter.	Length between webs.	Breadth.	Thickness fore and aft.			
(a)	*Baltimore*	135	42, 60, 94	42	15	7	19 H.P. 36 L.P.	15½	7	16	17	11¼	20¼	62 H.P. 79 L.P.	17⅜ H.P. 25⅞ L.P.
(b)	*Bancroft*	160	13½, 21, 31	20	6	3	5½	6	3	6	7½	4	2⅝	19½	4⅜
(c)	*Columbia*	160	42, 59, 92	42	16	7½	19⅝	17	8	21½	18	10¼	10	62	15⅛
(d)	*Concord and Bennington**	160	22, 31, 50	30	9	3	14	9	3	10	13¾	6½	7¼	42¼	10⅜
(e)	*Machias and Castine*	160	15¾, 22½, 35	24	7	3½	7	7	3½	7½	8	4¼	3½	23	5⅝
(f)	*Marblehead*	160	26½, 39, 63	26	10	5	12	10	5	15	13	6	6 1/16	39½	9 1/16
(g)	*Monterey*	160	27, 41, 64	30	11	4	12	11	4	15	14	7½	6	42	9¾
(h)	*Newark**	160	34, 52, 76	40	13	4	20	13	5	15	21½	10	12¼	64⅜	17¼
(k)	*New York* (forward engine)	160	32, 47, 72	42	13½	6	16	13½	6	17	15	9	8½	51½	12⅜
(l)	*Philadelphia**	160	38, 58, 86	40	14½	6½	23	15	7½	17	25	10½	13⅜	69¾	18⅞

Lengths are in inches. Pressures are in pounds per square inch.
* The engines in these ships have built crank shafts.

TABLE IV.—STRESSES DEDUCED FROM PREVIOUS DATA.

Line.	Ship.	I. Force on piston rod $= F_m$.	II. Augmented stress in aft L.P. main journal.	III. Augmented stress in L.P. crank pin.	IV. Stress in aft web from bending and turning moments.	V. Pressure on main bearing.	VI. Pressure on crank pin.
(a)	*Baltimore*	69.72	12,390	11,680	12,522	227 H.P. 120 L.P.	491
(b)	*Bancroft*	39.07	10,170	10,018	8,942	249	430
(c)	*Columbia*	80.19	10,500	9,292	12,501	235	383
(d)	*Concord and Bennington* *	43.56	11,260	11,770	6,976	163	432
(e)	*Machias and Castine*	34.98	9,907	9,703	11,421	216	380
(f)	*Marblehead*	87.14	11,203	12,002	11,114	257	388
(g)	*Monterey*	69.08	9,636	10,230	8,770	247	373
(h)	*Newark* *	61.02	12,760	13,723	6,658	197	496
(k)	*New York* (forward engine)	48.01	10,198	10,338	11,662	206	365
(l)	*Philadelphia* *	76.37	12,067	12,046	6,309	192	474

Column I.—The values are in pounds for a similar engine of one inch stroke.
Columns II to VI.—The values are in pounds per square inch. These quantities are constant for similar engines whatever the stroke.
* The engines in these ships have built crank shafts.

between the brasses plus the length of the aft bearing. This is, probably, fairer than taking the distance center to center where the forward bearing is the longer.

In Table IV is given the force F_m on the piston rod of a similar engine of one inch stroke; the stresses (augmented where shearing stress is present) in the metal of the aft main journal, crank pin and web; and the nominal surface pressures on the aft main bearing and crank pin. For the main journal this number is attained in the usual way, by dividing the total force on the journal by the product of the diameter of journal and length of brass. For the crank pin, the length between the crank webs was taken instead of the length of brass; this makes no material difference in the result. All these values are, of course, deduced from Table III by means of Tables I and II.

Table V gives the average and maximum values of stresses from Table IV. The last line of the former gives the constants which have been used in calculating the crank shafts represented by the curves Fig. VIII. These values in all cases lie between the average and maximum values given by the two previous lines of the table.

TABLE V.—AVERAGE AND MAXIMUM STRESSES FROM TABLE IV, AND THOSE USED IN CALCULATIONS.

Line.	Column.	I.	II.	III.	IV.	V.
(a)	Part of shaft.....	Aft L.P. main journal.	L.P. crank pin.	Aft web.	Aft main bearing.	Crank pin.
(b)	Stress.....	Augmented stress in metal.			Surface pressures.	
(c)	Average values, from Table IV.....	11,009	11,080	10,990*	210	421
(d)	Maximum values, from Table IV.....	12,760	13,723	12,522	257	496
(e)	Values adopted in calculations.....	11,500	11,500	11,500	220	440

*The figures for built cranks in Table IV are excluded in striking this average.

These values are all in pounds per square inch.

The three following assumptions were made to make the calculations definite:—

1st. That holes of one half the diameter of the corresponding parts were bored axially through the main journal and crank pin.

2d. That the breadth of the crank web is 10 per cent. greater than the diameter of the crank pin.

3d. That the clearance between the main bearing brass and the crank web is 5 per cent. of the main journal diameter.

The various dimensions of the shafts were calculated by trial and error. But it would be most tedious for the reader, and is unnecessary, to follow the work in detail or describe its arrangement.

As already explained, in Fig. VIII the horizontal scale is the value of F_m—the maximum force in pounds on the piston rod for a similar engine of one inch stroke. Each of the full line curves gives by its ordinate, read on the vertical scale, the dimension of the part noted at the end of the curve. This value, when multiplied by the actual stroke in inches, gives the dimension for the stroke s. The dotted curves will be explained in connection with the question of want of alignment of the bearings.

The curves are calculated from $F_m = 10$ to $F_m = 100$, which fully covers the limits of practice.

An example will clearly illustrate the use of the curves.

Example I. Engine cylinders = 26 inches, 40 inches, 60 inches.

Stroke = 30 inches.

Boiler pressure = 160 pounds per square inch.

From equation (1)

$$F_m = \frac{\pi\,(160 + 15)}{4 \times 30^2 \left(\frac{1}{26^2} + \frac{1}{40^2} + \frac{1}{60^2}\right)} = 64.11.$$

We at once have the following results from Fig. VIII.

Part of Shaft.	Ordinates from Fig. VIII.	Ordinates from Fig. VIII × stroke (= 30″).
Main journal, diameter	.345 inch.	10.35 inches.
length of bearing	.444 inch.	13.32 inches.
Crank pin, diameter	.350 inch.	10.50 inches.
length between webs	.413 inch.	12.39 inches.
Crank web, breadth	.385 inch.	11.55 inches.
thickness	.255 inch.	7.65 inches.
Span of main bearings center to center	1.402 inches.	42.06 inches.
Clear distance between bearings	.958 inch.	28.74 inches.
Clear distance between web and main bearing brass	.017 + inch.	.52 inch.

The curve giving the clear distance between the bearings is added to prevent confusion as to the meaning of "Span of main bearings center to center." This is really, as explained already, the actual distance from center to center on the supposition that the forward L.P. bearing has the same length as the after one.

Before treating the question of the want of alignment, let us examine the effect which very bad distribution of power among the three cylinders has on the stresses in the various parts of the shaft.

Increase of pressure in the L.P. receiver is the most common way in which inequality of steam distribution occurs at full power. It is also, in nearly all cases, the severest in straining the crank shaft. In Fig. III, if the L.P. power is increased, it is the first of the three maxima in the full line curve which is raised to form the principal maximum. Near that part of the revolution the reaction of the aft journal, and consequently the L.P. bending moment, is large; and is increased in a large ratio by the change in the distribution of power.

If the same increase of power occurs, instead, in one of the other cylinders, the total turning moment is affected in the same way, but it is one of the other two maxima which is then raised highest. About these maxima the L.P. bending moments are less important and are little affected by the change of steam distribution just supposed.

To be definite, let us take the very worst case which deserves any consideration. Suppose one cylinder to be doing half, another one-third, and the last one-sixth, of the total work. This is somewhat worse than the distribution in the *Castine's* engines at the full power trial. If we take the large power to be in the L.P., and the small power in the H.P., we may find the total turning moment approximately by adding 50 per cent. to the L.P. turning moment, Fig. III, and deducting 50 per cent. from that of the H.P. This procedure, though not correct, is quite near enough for our present purpose. We may similarly treat Fig. VII, and the same process may be applied when any of the other cylinders are giving excess of power.

After the curves in Figs. III and VII had been thus altered,

the moments were measured where maximum stress occurred, as in line (d) Table I, and similar tables constructed for each case, applying the same multipliers, except that line (f) was omitted, as the curves had been altered and the new tables constructed on definite assumptions as to the inequality of the steam stresses. The new moments were then applied to the shafts given by Fig. VIII, and the stresses calculated.

For brevity, only the results are shown in Table VI.

The order of cranks is H.P., L.P., I.P., as this gives rather larger bending moments than the reverse order.

Six such assumptions as to inequality of power in the three cylinders can be made, of which Table VI only gives three, viz: (a), (b) and (c); but a little consideration will show that the three not given would give smaller maxima than are given for the various parts, in Table VI.

Aft L.P. Main Journal.—The figures are given only for the case where the L.P. power is in excess—assumption (a)—as assumptions (b) and (c) would obviously give lower maxima. For all values of F_m the rise of stress over that given by Table I is nearly the same. It is not large and, as this part of the shaft is much less affected by want of alignment than the parts of the L.P. crank further forward, we need make no change on this account.

Aft L.P. Web.—Only the figures for assumption (a) are given, for the same reason as has just been noted above. The maximum stress cannot occur till the L.P. crank is about 36° past top center. The crank and connecting rod are then at a greater angle than when in the position taken in Table I. This largely accounts for the rise of stress being comparatively small. We must recollect that the stress in the webs has been estimated fully 10 per cent. too high, from an allowance for change of section (line (k), columns III and IV, Table I) being made, which was scarcely justifiable. To prevent confusion, this allowance was retained in constructing Table VI. Therefore, the figures 1.109, 1.096, 1.089, for the aft L.P. web, should be divided by 1.1. This reduces them to 1.008, 0.996, 0.990, respectively. Thus, at worst, the stress is just raised above the 11,500 pounds per square

TABLE VI.—CHANGES OF STRESS FROM VERY UNEQUAL DISTRIBUTION OF POWER AMONG CRANKS.

	Multiplier for stress from—			Part of crank shaft.		Aft L.P. main journal.	Aft L.P. web.	L.P. crank pin.	Forward L.P. web.	Forward L.P. main journal.
	H.P. cyl.	I.P. cyl.	L.P. cyl.	Position of L.P. crank.		45° past top center.	36° past top center.	36° past top center.	27° past top center.	18° past top center.
(a)	.50	1.00	1.50	Stress / 11,500	F_m= 10	1.147	1.109	1.008	.856	.655
					F_m= 50	1.155	1.096	1.052	.897	.814
					F_m=100	1.162	1.089	1.083	.925	.944
				Position of L.P crank.		.....	.....	At bottom center.	At bottom center.	At bottom center.
(b)	.50	1.50	1.00	Stress / 11,500	F_m= 10	.....	.....	1.064	1.018	.956
					F_m= 50	.....	.....	1.046	1.057	.983
					F_m=100	.....	.....	1.033	1.084	1.004
				Position of L.P. crank.		.....	.....	At bottom center.	At bottom center.	At bottom center.
(c)	1.00	1.50	.50	Stress / 11,500	F_m= 10	.....	.....	1.189	1.148	1.121
					F_m= 50	.....	.....	1.091	1.135	1.128
					F_m=100	.....	.....	1.024	1.128	1.131

inch allowed originally. No change therefore is required on this account in the shafts given by Fig. VIII.

L.P. Crank Pin.—The greatest rise of stress occurs here for assumption (c) and $F_m = 10$. This distribution of power—where the work done by the L.P. is very small—can hardly occur at full power, and I have no recollection of ever meeting it. When the valve gear is in full gear and the power is reduced by throttling, or by cutting off early in the H.P. alone, the large proportion of the work can readily be thrown on the H.P. crank, due to the lowering of the I.P. receiver pressure; but then the whole turning moment curve is lowered, and the crank shaft relieved from stress. We may then, I think, safely assume that to credit only one sixth of the whole work, at full power, to the L.P. crank, is a somewhat worse condition than will occur in practice or need be allowed for here.

The value for assumption (a), $F_m = 100$, is 1.083, but here again rise of stress is less rapid for want of alignment than in the forward L.P. web and main journal, and consequently a slight excess of stress in the pin may be allowed when the shaft is in line.

Forward L.P. Web.—If we reject assumption (c) as worse than will occur in practice, and divide the figures in this column by 1.1, we find in all cases the stress falling below the 11,500 pounds per square inch allowed for the maximum.

Forward L.P. Main Journal.—Here the stress rises to 11,500 pounds per square inch only in one case if assumption (c) is rejected.

On the whole then, we find that in no case is the maximum stress much raised by any bad distribution of power which will occur in reasonable practice, and that the parts most affected by want of alignment do not rise above the allowed maximum.

Assumption (a) raises the pressure on the L.P. crank pin to 600 pounds per square inch, and that on the main bearing to 282, multipliers being used as in Table II, except line (e).

Now we have to consider what effect want of alignment in the bearings has in increasing the stress in the shaft.

We have already seen that, when the shaft is in line, definite forces acting on the piston rod induce definite reactions from the main bearings. When the shaft is out of line, in addition to these first reactions there is a second set superimposed on them which depend on the elasticity of the shaft and the particular derangement of the bearings which may exist. These new reactions cause changes in the bending moments at each point of the shaft, and also in the turning moments of the crank pins and those parts of the webs lying outside the center line of the shaft. When the bearings are in line we get the portions of the shaft between the two aft bearings A and B, Fig. IV, most heavily stressed; consequently, those derangements of alignment of the bearings which most quickly add to the stress in this section of the shaft will most readily produce fracture.

If there is no bending moment at xx, Fig. IV, as has been assumed, want of alignment can only stress the shaft between A and B by affecting the reaction at the aft bearing A. The whole question, then, narrows itself down to this:—How much reaction will be induced at A by a given want of alignment of the bearings? I will give the answer to this question here, relegating its deduction to the end of the paper.

Let l, m, n, p, be the heights of the center line of the shaft at the points A, B, C, D, Figs. IV and VI, above a fixed straight base line, not shown. If instead of the actual shaft, Fig. IV, we have an originally straight bar, of diameter a, supported at points A, B, C, D, as in Fig. VI, then the reaction at A, due to want of alignment alone, is

$$-\frac{0.589 \times 10^6 \times a^4}{b^3}(4l - 9m + 6n - p). \quad . \quad . \quad (15)$$

So long, then, as the factor $4l - 9m + 6n - p$ retains the same value, the effect in stressing the aft portion of the shaft will be the same, no matter what the individual values of l, m, n, p, are. If the bearings are in line $l - m = m - n = n - p$, and the factor becomes zero, as it ought. We also see at once the relative importance of a given want of alignment in each bearing. Thus $\frac{1}{1000}$ inch vertical shift in bearing B would be as important as $\frac{9}{4000}$ inch in bearing A, or $\frac{9}{6000}$ inch in bearing C, and so on.

The expression (15), however, takes no account of the effect of the webs on the elasticity of the shaft, which is in most cases considerable. This may be approximately allowed for as follows. Suppose we cut off the crank shaft just forward of bearing B and aft of bearing A, Fig. IV, and let the connecting rod of the L.P. engine apply vertically, say, one pound to the crank pin when the crank is at top center. The shaft will be bent symmetrically about the center line of the L.P engine; and, supposing the pressures distributed equally over all bearing surfaces, we can readily calculate the angle at the center of the bearings A or B to which the center line of the shaft is deflected from its unstrained position. Now remove the webs, replacing them by circular parts of the same diameter as the main journals, and bring down the crank pin to the center of the shaft to make this new bar continuous. Again calculate the angle at A and B for one pound push on the crank pin. The ratio of the first to the second angle just calculated will give a factor representing approximately the increased elasticity of shaft produced by the webs. As increase of flexibility will decrease the reaction at A for a given value of $4l - 9m + 6n - p$, the expression (15) must be *divided* by the factor for the influence of the webs. In all figures given below the influence of the webs has been allowed for in this way.*

The point of the revolution at which the maximum stress will now occur will not be shifted from that found for the bearings in line in those parts of the shaft lying along the center line. For

*As before stated, after the work was practically complete, a new investigation of the distribution of pressure on the bearings was made which took more exact account of the influence of the webs. In the case of the bearings A, C and D in line, but B out of line, the pressure on A, then found, was in excess of the figures given by the earlier method by the following percentages:

$F_m =$ 10, 10.7 per cent.
$F_m =$ 20, 8.9 per cent.
$F_m =$ 60, 5.3 per cent.
$F_m =$ 100, 2.1 per cent.

If the effect of the turning moments in straining the crank pins and webs, referred to earlier, were taken account of, the whole value of expression (15) would be lowered a little, and thus in using it we err on the safe side.

the new bending moments due to want of alignment, merely form a constant addition to those previously found for the shaft in line, while the turning moments are not interfered with. Let us examine what happens in the case of the crank pin and, for simplicity, let us neglect the running clearance of the bearings. Suppose the bearing *B*, Fig. IV, to be slightly raised from its proper position, while the other three bearings remain in line. When steam is off, a downward reaction on the shaft will still exist at *A* due to the want of alignment. When under full steam the reaction at *A* for the L.P. at bottom center, instead of being *DE*, Fig. VII, as we found for the shaft in line, will be increased to, say, *DF*. All the reactions on the up stroke will be increased and those on the down stroke diminished by the amount *EF*. Thus at 36° past top center, the reaction instead of being *GH*, will be *GK*. In Fig. III the turning moment in the aft journal is given by the full line, and that on the crank pin with the shaft in line approximately by the dotted line. As we have seen, the difference of those two is due to the reaction at *A*. Error in the alignment of the bearings which will diminish this reaction will increase the turning moment in the crank pin and vice versa. Thus at 36° past top center, when the reaction is diminished to *GK*, Fig. VII, the crank pin turning moment will be increased from *GH* to about *GK*, Fig. III. If bearing *B* is shifted so far that the reaction *GH* is reduced to zero, the crank pin turning moment will increase to *GL*, and so on.

At bottom center the reaction at *A* does not affect the crank pin turning moment, which therefore retains its constant value *DE*, Fig. III. Thus, at bottom center, increase of reaction at *A* will always increase the stress on the crank pin, while at *G* any increase of stress will be due to increase of turning and decrease of bending moments, till the error of alignment actually reverses the pressure on bearing *A* at point *G*, when a further shift of bearing *B* will make both bending and turning moments at *G* increase. *D* is the position for maximum stress for the crank pin and forward web when the bearings are in line, and it will be found, as might be expected from what has just been said, that the maximum for these parts still occurs at this point

until the reaction at D is very largely increased due to error of the bearings. When $F_m = 10$—that is, when the bending moments are least important—the stress in the crank pin at point G would not be so great as at D, till the total reaction on bearing A at point D had become three times DE. The pressure on A would then always be in one direction and would vary between about $440 + 220 = 660$ pounds per square inch at point D, and $440 - 220 = 220$ pounds per square inch at point A. The pressure on bearing B, unless the brass was very long, would be much more intense even than this and, although the distribution of pressure over the bearings were good, it would not be possible to make them run cool. So we need only consider the effect of want of alignment on the crank pin when it is at point D. For the forward L.P. web, the error of the bearings would have to be even greater to make the maximum at G greater than at D.

When the shaft is in line, we have seen that the aft web is most heavily stressed near the main journal. With the shaft out of line so far that there is no reaction at bearing A, there will of course be no bending moment on the aft web, and the turning moment it transmits will no longer be smaller at the crank pin than at the main journal end. Thus the stress all over the web will be the same. From this it at once follows that the stress in the web varies more quickly for a given want of alignment at the crank pin than at the main journal end; and, when the reaction at A is actually reversed through error in the bearings, the former point will be the more heavily stressed. But it has not been considered necessary to give, in Table VII, figures for more than the inner end; as, when the shaft is considerably out of line, the L.P. forward web is the more heavily stressed of the two.

Table VII exhibits the effect of the bearing B being out of line by $\frac{1}{1000}$ of an inch, when the engine is reduced to a similar one of one inch stroke. This, of course, corresponds to a want of alignment of B in a 20-inch stroke engine of

$$\frac{20}{1000} = \frac{1}{50} \text{ inch.}$$

In expression (15) we have $l = n = p = 0$, and $m = 0.001$. (15) then becomes $\frac{5301\ a^4}{b^3}$.

Before entering the results in Table VII, line (a), they have been divided by the factor for the influence of the webs, explained above, and by the area of the journal from Fig. VIII. Line (a), then, gives the actual increase of reaction at bearing A per square inch supposing it uniformly distributed.

By adding this new reaction at A to that already found for the shaft in line we may find the bending and turning moments now induced in the various parts of the shaft, and thus calculate the maximum stress.

TABLE VII.—EFFECT OF BEARING B BEING 0.001 INCH OUT OF LINE.

Other bearings supposed in line. Stroke of engine = 1 inch.

Line.		F_m.	10	20	40	60	80	100
(a)		Increased reaction at bearing A	560	345	203	148	117	98
(b)	Increase of maximum stress.	Aft L.P. main journal...	1,090	1,033	936	876	827	789
(c)		Aft L.P. web.....	2,987	2,509	2,015	1,743	1,563	1,430
(d)		L.P. crank pin.....	4,313	3,710	3,044	2,681	2,424	2,231
(e)		Forward L P. web.....	9,454	7,887	6,330	5,456	4,870	4,442
(f)		Forward L.P. main journal.....	7,350	6,318	5,227	4,649	4,254	3,956
(g)	Maximum stress when all bearings are in line.	Aft L.P. journal and crank pin.....	11,500					
(h)		Aft L.P. web.....	11,500 ÷ 1.1 = 10,455					
(k)		Forward L.P. web.....	9,819	9,964	10,182	10,318	10,446	10,553
(l)		Forward L.P. main journal.....	10,037	10,137	10,263	10,372	10,457	10,540

The greatest stress in each part for the shaft in line is given in lines (g) to (l), Table VII. In lines (h) and (k) no influence is ascribed to change of section in raising the stress in the webs. By deducting the values in lines (g) to (l) from the new maximum stress found when bearing B is 0.001 inch out of line we find the rise of stress for the several parts, entered in lines (b) to (f). The rise of stress will in all cases be nearly propor-

tional * to the want of alignment of bearing B. And as, by means of expression (15), we can find the error of alignment of bearing B which is equivalent to any other derangement of the four bearings, the effect of want of alignment can in any case be at once calculated from Table VII.

The principal indication an engineer has when the engine is running that the shaft is out of line, is the heating of the bearings. In a very careful set of experiments on friction by Mr.

* For the inner ends of the webs the increase of stress will be exactly proportional to the want of alignment of bearing B, as only the bending moments are involved in the question. In the round parts of the shaft we have—equation (8)—

$$T_a = B + \sqrt{\left[B^2 + \left(\frac{10\ T}{7}\right)^2\right]}.$$

This gives for the rate of increase of augmented moment per unit increase of bending moment

$$\frac{d\ T_a}{d\ B} = 1 + \frac{B}{\sqrt{\left[B^2 + \left(\frac{10\ T}{7}\right)^2\right]}}.$$

The greatest deviation from constancy in this rate within the limits of stress with which we have to deal will be in the case of the forward L.P. main journal, when $F_m = 100$. When the shaft is in line we will have

$$\frac{d\ T_a}{d\ B} = 1.207.$$

When the shaft is so far out of line that there is no reaction at bearing B, or when bearing B is removed—a case to be treated immediately—we get

$$\frac{d\ T_a}{d\ B} = 1.518.$$

As $1.518 + 1.207 = 1.258$, we may consider that there is, in the limits of practice, a variation of the rate of increase of stress per unit change of bending moment, or reaction at bearing A, of somewhat over 25.8 per cent. For $F_m = 10$ this figure becomes 11.8 per cent. These changes of rate are not important, as no very exact deduction is to be made from Table VII; but, instead of directly calculating the stress for B = 0.001 inch out of true, the want of alignment which would raise the augmented stress to 12,650 pounds—that is 10 per cent. above 11,500 pounds originally allowed—was calculated, and from that the rate per 0.001 inch displacement of B deduced. The figures in lines (b), (d) and (f), Table VII. may therefore be looked on as mean values over the range of variation which usually occurs in practice. If calculated exactly, the figure 20,753, for $F_m = 100$, line (c) of Table VIII, would have been 22,690; thus, even in this very extreme case, the error is comparatively small, and Table VII may be taken as practically correct.

Deviation from exact proportionality of rise of stress to error of bearing B is much less for the aft L.P. main journal and crank pin than for the forward main journal.

Beauchamp Tower (see Trans. Inst. M. E., London, 1883 and 1885) trials were made to find the greatest load which his journal would carry. The journal was four inches diameter by six inches long; and the brass, which was very carefully fitted, embraced nearly a semicircumference. Oil bath lubrication was used and the pressure was steady. He found that with rape oil the brass seized with a nominal pressure of 573 pounds per square inch, and with mineral oil at 625 pounds per square inch. The brass in each case was taken out and rescraped, but no better result was got on repeating the trials.

Let us now see what stresses might come on the shaft before seizing would occur under the most favorable conditions for the bearings running cool. Lubrication would not often be nearly so perfect as that of the oil bath in Mr. Tower's experiments but, on the detection of a tendency to heat, it would become copious, and we must consider that the journal is very freely oiled.

If there is a single bearing at B, it will not often be longer than about one and a half times the length that has been given to bearing A. (Even if longer than this the pressure from the bent shaft will distribute itself so much worse as to make it scarcely more effective than the shorter bearing.) The reaction at B, *when the shaft is in line*, also is fully one and a half times that at bearing A. So we may count this part of the reaction to amount to at least 220 pounds per square inch. Suppose, to be definite, that bearing B is raised—though the particular assumption made matters little. *With steam off*, the first effect will be to raise the shaft from bearings A and C. When it has been raised to the extent of the running clearance, a downward pressure on the shaft will be exerted by the caps of bearings A and C which will increase as B is further raised. The reaction *thus* caused at B will, however, be about twice that at A; and, allowing for B being longer than A, the mean pressure at B will be about one and a half times that at A. Even with the shaft in line the pressure will by no means be so well distributed over the bearing as it was found to be in Mr. Tower's experiments. For, the continually varying flexure of the shaft will cause a very slight rounding of the brasses in a longitudinal plane and, consequently, the fall of

pressure on each side of the point of maximum pressure at any time will be much quicker than for a bearing where there is no changing flexure due to variable bending moment. That is, in the former case, the endward escape of oil will be so much easier than in the latter.

With the shaft out of line as supposed, the distribution of pressure would generally become much worse. But even if such were not the case, we could not without seizing attain the same values of nominal pressure as Mr. Tower did. Let us suppose that under very favorable circumstances the maximum value of the nominal pressure on bearing B could be run up to 600 pounds per square inch without seizing. This gives for the reaction due to want of alignment alone $600 - 220 = 380$ pounds per square inch. This is also, obviously, the mean nominal pressure over the revolution very nearly, and is only a little less than two thirds of the limiting pressure found by Mr. Tower under immensely better conditions. As the reaction due to want of alignment is now greater than that due to steam, the pressure on B is no longer intermittent, and thus the access of oil to the part of the bearing under pressure is less free.

To suppose two bearings at B would be a worse assumption than that above. For the longer the length of shaft which the bearings are spread over, the more will its flexibility reproduce very local reactions.

To produce 380 pounds per square inch rise of pressure at B will, according to the approximate supposition made above, require a rise of $380 \div 1.5 = 253$ pounds per square inch at A. From Table VII, when $F_m = 10$, a rise of 560 pounds per square inch at bearing A increases the stress in the forward L.P. main journal from 10,037 pounds to $10{,}037 + 7{,}350$. Thus a rise of pressure at A of 253 pounds will cause a stress in this part of $10{,}037 + \frac{253}{560} \times 7{,}350 = 13{,}358$ pounds. Altering lines (d), (e) and (f), Table VII, in this way, we get the figures in Table VIII. The corresponding results from lines (b) and (c), Table VII, are not given, as the stresses would in all cases be smaller than for the crank pin and parts forward, which are given.

The stress in the forward L.P. main journal is measured just at the junction with the forward web and the same allowance of 10 per cent. made for change of section as formerly. As the total reaction from bearing A now rises above that on the crank pin, the moment which the forward L.P. main journal has to withstand is greatest near the center of bearing B, but if the 10 per cent. allowance for change of section is correct, the greatest stress will still occur at the junction with the web.

It will be found on trial that even in this extreme case the shearing stress due to torsion of the webs has no effect on the maximum. Indeed, it would not have, even if the turning moment and the moments from the pressures on bearing A and the crank pin remained constant as the shaft revolved.

In line (d), Table VIII, is entered the error of bearing per one inch stroke of engine to which the stresses in the previous lines correspond.

The questions now arise: What factor of safety remains under the conditions of Table VIII, and what alterations are required in the curves to make the factor of safety sufficient in all cases?

For steel of 60,000 pounds tensile strength the elastic limit will be in the neighborhood of 30,000 to 36,000 pounds.

TABLE VIII.—STRESSES IN SHAFTS UNDER EXTREME CONDITIONS OF WANT OF ALIGNMENT OF BEARINGS.

Line.	F_m.	10	20	40	60	80	100
(a)	Maximum stress in L.P. crank pin.....	13,449	14,220	15,294	16,083	16,742	17,259
(b)	Maximum stress in forward L.P. web	14,090	15,748	18,071	19,644	20,976	22,021
(c)	Maximum stress in forward L.P. main journal	13,358	14,770	16,777	18,319	19,656	20,753
(d)	Corresponding error of bearing B.....	0.452	0.773	1.246	1.710	2.163	2.581

Stresses are augmented and in pounds per square inch.
Error of bearing B is in thousandths of an inch per one inch stroke of engine.

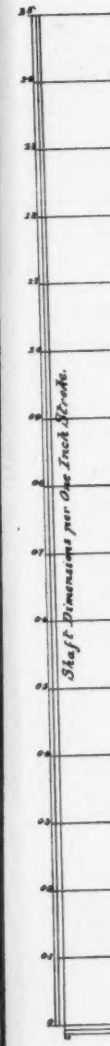
The factor of safety for the web calculated from the elastic limit, taken at say 33,000, is $\frac{33,000}{22,021} = 1.5$. If the shaft is sound

this should not overstrain it, but it leaves little margin for imperfections or additional stresses due to shocks, &c. As F_m becomes smaller, the factor of safety rises. It is important to note that the edge of the web which at bottom stroke has a tension (or compression) of 22,021 pounds, has at top stroke a a compression (or tension) of only 2,723 pounds. This can easily be verified. Thus there is no complete reversal of the stress, which Wöhler found so readily produced fracture; and thus, probably, the factor 1.5 is very nearly a true estimate. Even reversal of the engine will not produce complete reversal of the above stresses, for when the engine is running ahead one pair of opposite edges of the webs have to withstand the maximum stress, and when running astern it is the other pair.

With the forward L.P. journal the case is somewhat different. We have seen, as given earlier in a footnote, that the true figure for the augmented stress when $F_m = 100$ is 22,690 pounds instead of that given in Table VIII. Shearing stress due to turning moment produces little of this, and calculating the principal tension and compression we find:—

Line.	Position of L.P. crank.	At bottom center.	At top center.
(a)	Principal Stresses.....	Tension, 21,163 pounds.	Compression, 13,646 pounds.
(b)		Compression, 1,688 pounds.	Tension, 1,409 pounds.
(c)	Stress parallel to axis of shaft....	Tension, 19.476.	Compression, 12,237.
(d)	Angle between axis of shaft and principal stress in line (a) above..	15° 46′.	—(17° 49′).

The figures in lines (a), (b) and (d) are, of course, deduced by means of equations (5), (6) and (4), and the stresses are in no way augmented. Line (c) gives the stresses induced by the bending moments alone. If the words "tension" and "compression" are interchanged, the above statement holds good for the diametrically opposite point of the shaft in which the highest stress is a compression.



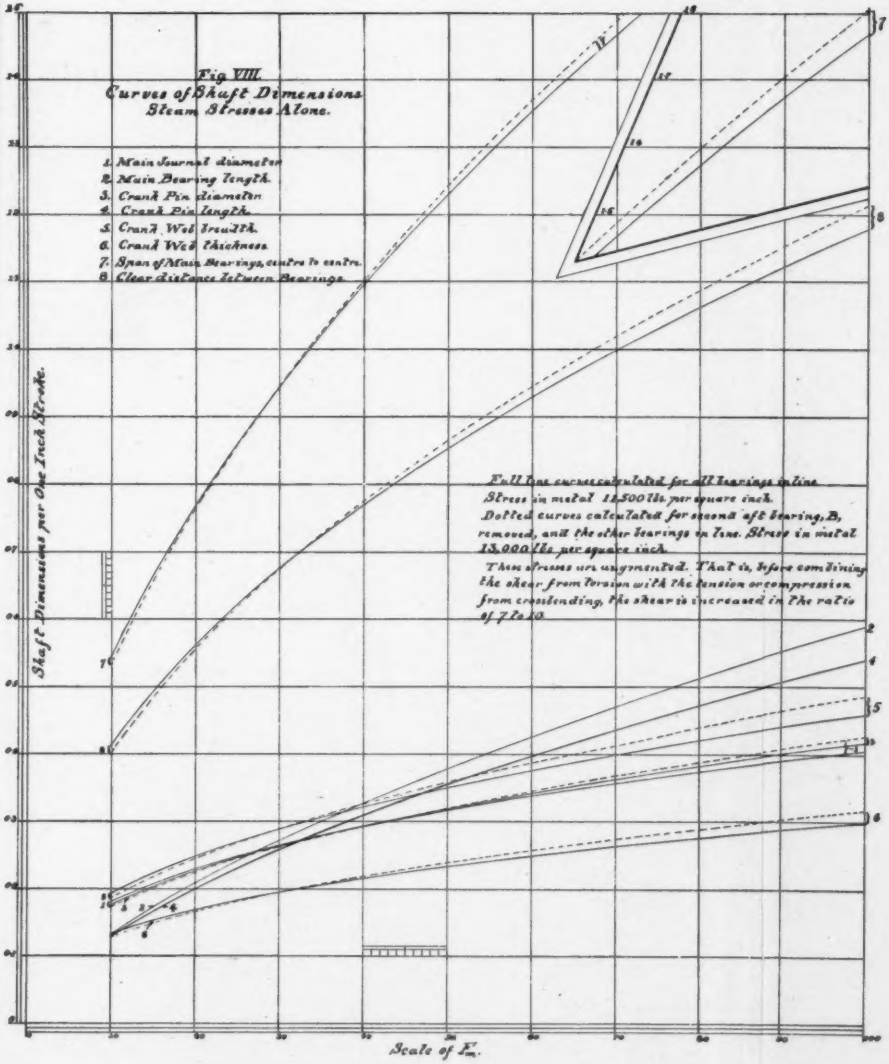
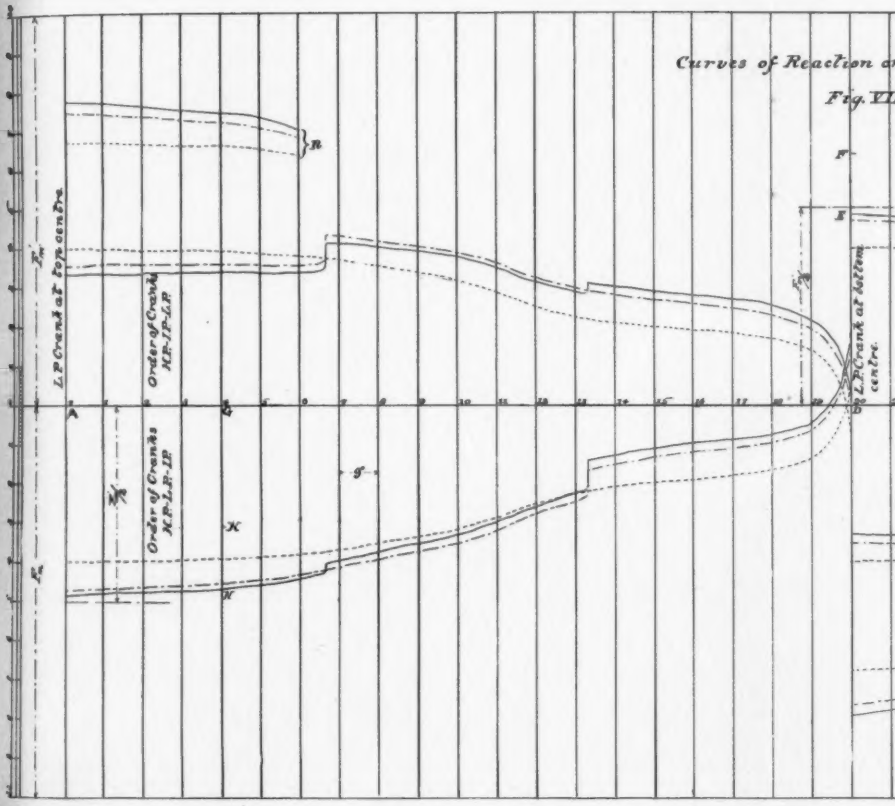
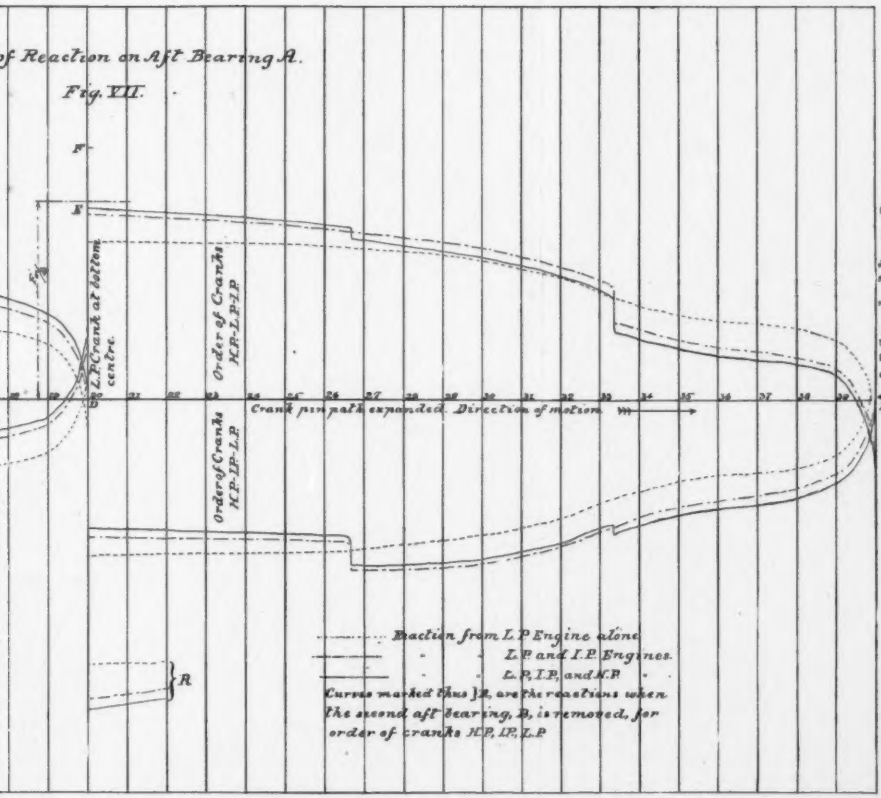


Fig VIII.
Curves of Shaft Dimensions
Steam Stresses Alone.
1. Main Journal diameter
2. Main Bearing length.
3. Crank Pin diameter
4. Crank Pin length.
5. Crank Web breadth.
6. Crank Web thickness.
7. Span of Main Bearings, centre to centre.
8. Clear distance between Bearings.
Shaft Dimensions per One Inch Stroke.
Full line curves calculated for all Bearings in line.
Stress in metal 11,500 lbs per square inch.
Dotted curves calculated for second aft Bearing, B, removed, and the other Bearings in line. Stress in metal 13,000 lbs per square inch.
Thus stresses are augmented. That is, before combining the shear from torsion with the tension or compression from crosslanding, the shear is increased in the ratio of 7 to 10.
Scale of Ins.


Curves of Reaction
Fig. VI
L.P. Crank at top centre
Order of Cranks
K-I.P.-L.P.
Order of Cranks
K-L.P.-I.P.
L.P. Crank at bottom centre
A
B
C
D
E
F
G
H
K
L
M
N


of Reaction on Aft Bearing A.
Fig. VII.
F
E
L.P. Crank at bottom centre.
Order of Cranks H.P-L.P-I.P.
Order of Cranks H.P-I.P-L.P.
Crank pin path expanded. Direction of motion
L.P Crank at top centre.
Reaction from L.P Engine alone
L.P and I.P. Engines.
L.P, I.P, and H.P
R
Curves marked thus]R, are the reactions when the second aft bearing, B, is removed, for order of cranks H.P, I.P, L.P

We see, then, that while the stress does not rise quite so high as for the web, the range of stress is considerably greater, though the directions of the alternate maximum tension and compression are inclined at an angle of $15° 46' + 17° 49' = 33° 35'$. If we neglect the shearing stress altogether, line (c) shows that we have still considerably greater range of stress than for the web; and, though it cannot be estimated exactly, I think we may conclude that the factor of safety is in this case somewhat smaller for the journal than for the web.

I will quote one experiment of Wöhler's,* which it is instructive to compare with what has just been said. The steel experimented on was cut from a bar supplied by Vickers, Sons & Co., in 1868, and gave the following results in a tensile test:—

Breaking stress per square inch of original section, 65,270 lbs.
Breaking stress per square inch fractured section, 115,988 lbs.
Percentage of elongation before fracture, ... 19.5

As the length measured for elongation is not mentioned in "Engineering" no conclusion can be drawn from this figure. Judging from the contraction of area, the steel was probably somewhat hard.

Round bars of this material were fixed horizontally in a testing machine and rotated. A constant vertical pull was applied at the outer end, so that the bar was submitted to cross bending, and thus the stress at each point exactly reversed each half revolution. Several bars were experimented on with the following results:—

Maximum stress in fibers in pounds per square inch.	*Number of revolutions of bar.*
36,380	51,240 bar broken.
34,240	72,940 " "
32,100	205,800 " "
29,960	278,740 " "
27,820	564,900 " "
25,680	3,275,860 " "
23,540	8,660,000 bar not broken.

* See "Engineering," Vol. XI, pp. 244 and 349.

Wöhler's experiments showed that under repetition of stress the range as well as the maximum stress was an important factor in determining the fracture of the bar. In the last trial above (where the bar was not broken) the stress ranged from 23,450 pounds tension to 23,450 pounds compression—that is, through 46,900 pounds. If the greatest compression had been less than 23,450 pounds, the greatest tension could have been increased over this figure; till, had the stress been purely a tensional one, varying between a maximum and zero, the upper limit could have been made about 40,000 pounds with equal safety. Consideration of this experiment seems to support what has been said with regard to the factors of safety of the forward L.P. web and main journal.

$F_m = 100$ lies beyond the limits of practice, and heating of the bearings is almost sure to occur much earlier than has been supposed, through very bad distribution of pressure over one or other of the bearings when these are out of line and the caps kept screwed down. Thus the figures in Table VIII certainly represent the upper limit of what may possibly occur. If then, the shafts shown in Fig. VIII are built of 60,000 pounds steel, and are perfectly sound, the following conclusions seem warranted by all that has gone before :—

By no reasonable inequality in the distribution of power, and by no conditions arising from want of alignment of the bearings under which the journals can be made to run cool, will the shafts be fractured, or even overstrained. The factor of safety for the larger values of F_m will however be small. With 80,000 pounds steel, of which the crank shafts for the smaller vessels of the U. S. Navy are usually made, the factor of safety would in all cases be fairly satisfactory even under such extreme conditions.

There is another limitation on the occurrence of these very high stresses in the crank shaft. Take for example the engine of the *Columbia*. From Table IV we find $F_m = 80.19$, and the stroke is 42 inches. From Table VIII, line (d), under $F_m = 80$ we find that the error of alignment of bearing B which would set up the stresses in the shaft given in the previous lines of the same table, is 0.002163 inch per one inch stroke. For 42 inches

stroke the error would be $0.002163 \times 42 = 0.091$ inch. Adding to this about 0.01 inch for the running clearance gives us practically one tenth of an inch. This is a quantity which would lie a long way outside of the most careless practice. Thus, for the larger values of F_m, there ought to be little danger of the high stresses given by Table VIII being approached unless in the case of short stroke engines, and these will usually have shafts of very strong steel.

Following Mr. Milton's procedure in the papers already referred to*, we may find the stress in the shaft when one of the bearings is removed. This is a condition to which every shaft is certain to be submitted occasionally, through slacking back of the bearing cap when the journal is heating. In most cases the deflection of the shaft is so small that a very slight easing of the cap will remove all pressure. The worst case is to suppose bearing B removed, except in the case of the forward L.P. main journal, where removal of bearing A or bearing B would produce nearly the same stress.

We must proceed as before to calculate the reaction f at A, supposing all influence from bearing B, in Figs. IV and VI, removed.

For B_c we now get the expression,

$$B_c = \frac{b}{32}(2F_1 + 7F_2 + 5F_3).$$

And taking moments about C we find

$$f = \tfrac{1}{64}(2F_1 - 9F_2 - 43F_3). \quad . \quad . \quad . \quad (16)$$

As before, the reaction f has been laid down in a curve: only the portions required are shown in Fig. VII, and are marked thus $\}R$.

The turning moments for the main journals shown by Fig. III are, of course, not altered. But as the reaction f at bearing A is increased, the turning moment on the L.P. crank pin is lowered, except at points A and D, when the L.P. crank is at top or bot-

* See *Transactions* of the Institution of Naval Architects, vols. XX and XXII.

tom center. . Obviously, the maximum stress will still occur at about the same points of the revolution as before, but the heaviest bending moments will be for the order of cranks H.P., I.P. and L.P.

The arms for the bending moments on the forward L.P. main journal, arising from the reactions on bearing A and the L.P. crank pin, are called H and H_1, respectively; and those for the bending moments on the forward L.P. web, arising from the same reactions, K and K_1; as shown in Fig. IV.

Table IX gives the deduction of the bending and turning moments now taken in the calculations. The construction of the table will be understood from what has been said with regard to Table I. Line (g) of Table I—"Increase for errors of adjustment"—is omitted here, as removal of bearing B will bring the center of pressure toward the inside of bearing A, and slightly reduce the arms of the bending moments. In line (d), column VI, the factor 0.45 in Table IX, instead of being $0.5b$, as in Table I, makes the ten per cent. reduction of moment due to the distribution of pressure over the crank pin. In columns V, VII and X, of Table IX, no increase is made for unequal steam stresses, as already explained in connection with the crank pin, Table I. No allowance is made for change of section in the webs.

Applying these moments to the shafts given by Fig. VIII will induce the stresses given by Table X. In line (f) the deflection at bearing B is given in thousandths of an inch per one inch stroke, the allowance explained above being made for the influence of the webs.

If the whole investigation had proceeded on the assumption of bearing B being removed, the constants deduced from the various engines would have been higher than those given in Table IV, except in the case of the total force on the crank pin. Taking the mean value of the stresses in lines (c), (d) and (e), Table X (the most highly stressed parts of the shaft) we find that it is between 13,000 and 14,000 pounds per square inch. This mean will be practically the same as we would have found had we deduced this constant from the ten engines of the U. S. Navy, in the same way as those of Tables IV and V are obtained.

TABLE IX.—TURNING AND BENDING MOMENTS WITH BEARING *B* REMOVED.

Line.	Column.	I.	II.	III.	IV.	V.	VI.	VII.	VIII.	IX.	X.	XI.	XII.
(a)	Part of crank shaft	Aft L.P. main journal.		Aft L.P. crank web.		L.P. crank pin.		Forward L.P. crank web.			Forward L.P. main journal.		
(b)	Moment	Turning.	Bending.	Turning.	Bending.	Turning.	Bending.	Turning.	Bending.		Turning.	Bending.	
(c)	Point of revolution at which moment is measured	L.P. crank about 36° past top center.				L.P. crank at bottom.							
(d)	Value of moment from Figs. III and VII.	$.85\ F_m$	$.74\ F_m\ h$	$.808\ F_m$	$.75 \times .9\ F_m\ k = .675\ F_m\ k$	$.77\ F_m$	$.79 \times .45\ b\ F_m$	$.77\ F_m$	$.79\ F_m\ K - F_m\ K_1$		$.77\ F_m$	$.79\ F_m\ H - F_m\ H_1$	
(e)	Multipliers. Ten per cent. loss from boiler pressure to absolute zero	.9	.9	.9	.9	.9	.9	.9	.9	.9	.9	.9	.9
(f)	Increase for unequal steam stresses	1.06	1.1	1.06	1.1	.....	1.1	.....	1.1	1.1	.....	1.1	1.1
(h)	Shearing stress augmented in ratio 7 to 10	10/7	.....	.....	.....	10/7	.....	.....	.....	.....	10/7	.....	.....
(k)	Increase for change of section	1.05	1.1	.....	.....	.....	.....	.....	.....	.....	1.05	1.1	1.1
(l)	Value of moment taken in calculation	$1.216\ F_m$	$.806\ F_m\ h$	$.771\ F_m$	$.668\ F_m\ k$	F_m	$.352\ F_m\ b$	$.693\ F_m$	$(.782\ K - .99\ K_1)\ F_m$		$1.04\ F_m$	$(.860\ H - 1.089\ H_1)\ F_m$	

The dimensions of the shafts given by the dotted lines in Fig. VIII are increased so as to give a stress in the crank pin, forward web and forward journal, of 13,000 pounds per square inch, when bearing B is removed. This was done by dividing the stresses in Table X by 13,000, and taking the cube root of the ratio thus found. The ordinates of the curves for the journal and pin diameter, and web breadth and thickness, were then simply increased in this proportion; no notice being taken of the slight alteration of b consequent on the increased thickness of the crank webs. Only one curve is given for the crank pin and main journal diameter. This is really the curve for the main journal diameter, and that for the crank pin would run below but exceedingly close to it.

TABLE X.—STRESSES IN SHAFTS, FIG. VIII, WHEN BEARING B IS REMOVED.

Line.	F_m.	10	20	40	60	80	100
(a)	Maximum stress in aft L.P. journal.....	11,693	11,792	11,947	12,080	12,190	12,288
(b)	Maximum stress in aft L.P. web.....	11,000	11,177	11,450	11,620	11,780	11,912
(c)	Maximum stress in L.P. crank pin.....	12,146	12,405	12,782	13,059	13,286	13,484
(d)	Maximum stress in forward L.P. web.....	11,632	12,430	13,534	14,288	14,916	15,440
(e)	Maximum stress in forward L.P. journal.....	11,400	12,066	13,062	13,867	14,554	15,158
(f)	Maximum deflection at bearing B.....	0.293	0.476	0.807	1.110	1.398	1.678

Stresses are in pounds per square inch.

Deflections are in thousandths of an inch per one inch stroke of engine.

The average pressure on the crank pin is still 440 pounds per square inch, but due to the increased flexure of the shaft the bearing will not be so good.

The average pressure on bearing A rises to 327 pounds per square inch.

In each case of a shaft running with bearings slightly out of line the stresses can be practically reduced to this case of one bearing removed by slacking back one or more of the main bearing caps. The condition only occurs occasionally, but line (f), Table X, shows that it will present itself much more readily in the case of the smaller than of the larger values of F_m; and for a proportionately smaller error of bearing in a short stroke than in a long stroke engine.

After careful consideration it was determined to place all the above evidence before the reader so that each individual designer might be in a position to judge for himself how much allowance he would make for the occasional occurrence of the stresses in Table X, and the possible occurrence of the higher stresses in Table VIII, rather than make a definite assumption which, at best, would only be the expression of one individual opinion, and could not satisfactorily be made to cover all cases.

My own judgment is—For the smaller values of F_m, where the dotted curves are lower than the full curves, the full curves should be used. For higher values of F_m, where the deflection at bearing B (got by multiplying the deflection from line (f), Table X, by the stroke) is about $\frac{1}{40}$th of an inch, the dotted curves should be used. For engines of longer stroke, a value between the full and dotted curves might safely be adopted. In an engine of 60 inches stroke, with $F_m = 80$, the deflection at bearing B from Table X $= 0.001398 \times 60 = 0.084$ inch. The bearings should, with care, never get out of line to more than a small fraction of this amount, nor need the bearing cap be slacked back nearly so far to allow freer lubrication. Therefore, if the forging can be perfectly depended on, the full line curve might be used here.

If the deflection at bearing B is less than $\frac{1}{40}$th inch, it would be well to make some allowance over the dimensions given by the dotted curves, Fig. VIII, unless steel stronger than 60,000 pounds per square inch is used. However short the stroke, the excess of dimensions over the full line curves Fig. VIII need, I think, never exceed twice the difference of the dotted and full curves. This, under the extreme condition of Table VIII, will in all cases leave a factor of about 2 from the elastic limit for steel of 60,000 pounds tensile strength. This additional strength in small engines is required not only because a very small error of adjustment will produce these high stresses; but the bearings do not heat so readily as in a larger engine, no doubt on account of the greater readiness with which the heat generated escapes, and thus warning that the shaft is under heavy stress will not be so soon given. Again, the larger engine has usually the greater number of attendants, though the number of bearings to be

looked after is nearly the same in both cases. For this and other reasons the smaller engine is often less carefully watched.

Fig. VIII gives definite values for the dimensions of the several parts of the shaft, but it will frequently be desirable to alter some of these to accommodate other parts of the engine design. It is necessary, then, to find what changes will have to be made in the other parts of the shaft to keep the strength unaltered. If, for instance, the span of the bearings is altered, the arms for the various bending moments will also be altered, while the turning moments in the journals will remain unchanged. The reactions of the main bearings will also, generally, alter slightly, but this we may neglect.

The change of stress induced by a given change in the arm of any of the bending moments will be greater for the shaft out of line than for perfect alignment as, in the former case, the bending moment is larger; but, as any change required in the shaft dimensions is always small, it will be sufficient to give the corrections only for the case of bearing B removed.

Crank Pin.—Suppose the span of main bearings, b, Fig. IV, increased to $b + \delta b$, so that the augmented moment on the crank pin is changed from

$$.352b + \sqrt{\{1 + .352^2b^2\}}, \text{ (See Table IX.)}$$

to

$$.352\,(b + \delta b) + \sqrt{\{1 + .352^2\,(b + \delta b)^2\}}.$$

Let the corresponding change of crank pin diameter be from γ to $\gamma + \delta\gamma$.

Then we have the proportion

$$\gamma^3 : (\gamma + \delta\gamma)^3 :: .352b + \sqrt{(1 + .352^2b^2)} : .352\,(b + \delta b) + \sqrt{\{1 + .352^2\,(b + \delta b)^2\}}.$$

It will be found in all cases sufficiently approximate to expand the quantities under the radical sign and retain only two terms.

Then, dividing the first two terms by γ^3 we get

$$1 : 1 + \frac{3\,\delta\gamma}{\gamma} :: 1 + .352\,b + \frac{.352^2\,b^2}{2} : 1 + .352\,b + \frac{.352^2\,b^2}{2} + .352\,\delta b + .352^2\,b\,\delta b.$$

By a well known transformation

$$1 : \frac{3\,\delta\gamma}{\gamma} :: 1 + .352\,b + \frac{.352^2\,b^2}{2} : .352\,\delta b\,(1 + .352\,b),$$

or
$$\delta\gamma = .117\,\gamma\,\delta b \times \frac{1 + .352\,b}{1 + .352 + \frac{.352^2\,b^2}{2}}.$$

When $F_m = 100$, $b = 1.804$, and

$$\frac{1 + .352\,b}{1 + .352\,b + \frac{.352^2\,b^2}{2}} = .890.$$

Thus we may write, with ample accuracy in all cases,

$$\delta\gamma = .117\,\gamma\,\delta b. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

For stroke s we have,

$$\delta\gamma' = \frac{.117\,\gamma'\,\delta b'}{s}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

Main Journal.—The corresponding expressions for the addition δa to the main journal diameter are—

For one inch stroke,

$$\delta a = .276\,(\delta H - 1.266\,\delta H_1)\,a. \quad . \quad . \quad . \quad (20)$$

For stroke s,

$$\delta a' = .276\,(\delta H' - 1.266\,\delta H_1)\,\frac{a'}{s}. \quad . \quad . \quad . \quad (21)$$

This is deduced from the moments in columns X, XI and XII, of Table IX; as, within the limits of practice, the stress is greater on the forward than on the aft L.P. main journal.

Crank Web.—The stress due to turning moment in the forward L.P. crank web is

$$\frac{6 \times .693\,F_m}{\varepsilon^2\,e} \text{ (see Table IX)}$$

where e is the fore-and-aft thickness, and ε the breadth, as shown in Figs. IV and V.

That due to bending is

$$\frac{.782\,K - .99\,K_1}{e^2\,\varepsilon} \times 6\,F_m, \text{ (See Table IX).}$$

As the total stress is 13,000 pounds, we have

$$\frac{6 \times .693}{\varepsilon^2 e} F_m + \frac{.782\,K - .99\,K_1}{\varepsilon\, e^2} \times 6\,F_m = 13{,}000,$$

or

$$\frac{F_m}{\varepsilon\, e}\left(\frac{1}{\varepsilon} + \frac{1.128\,K - 1.429\,K_1}{e}\right) = 3{,}127. \quad . \quad . \quad (22)$$

For stroke s

$$\frac{F_m s^2}{\varepsilon' e'}\left(\frac{s}{\varepsilon'} - \frac{1.128\,K - 1.429\,K_1}{e}\right) = 3{,}129. \quad . \quad . \quad (23)$$

If any of the quantities ε', e', K' or K'_1, are altered, one or more of the others (as the necessities of the design dictate) should also be altered to keep equation (23) true. This can be done very readily.

As the areas of the main journal and crank pin, given by Fig. VIII, correspond to pressures of 220 and 440 pounds per square inch, respectively, with all bearings in line (the pressure for the main bearing A rising to 327 pounds if bearing B is removed), the area required for any desired pressure can at once be calculated.

Example II.

To illustrate the use of equations (18) to (23), suppose that the aft main bearing in *Example I*, had to be moved 5 inches aft to accommodate, say, an eccentric between the bearing and aft web; and that it was desired to reduce the surface pressure in the main bearing to 180 pounds per square inch for all bearings in line. As the diameter of the shaft will be but slightly affected, the required reduction of bearing pressure will be made by increasing the length of the bearing to

$$\frac{13.32 \times 220}{180} = 16.28.$$

The determination of dimensions in *Example I* was made from the full line curves, Fig. VIII, though for 30 inches stroke probably most designers would choose values nearer those given by the dotted curves. The figures already given are however retained here, as this example would not be made clearer by changing them.

Crank Pin.—b' will be increased from 42.06 to

$$42.06 + 5 + 16.28 - 13.32 = 50.02.$$

Thus $\delta b' = 50.02 - 42.06 = 7.96$.

Thus from equation (19) we get the required increase of crank pin diameter,

$$\delta \gamma' = \frac{.117 \times 10.50 \times 7.96}{30} = .33,$$

and the new diameter of the crank pin is

$$10.50 + .33 = 10.83 \text{ inches.}$$

Main Journal.—We have $\delta H' = 7.96$ and $\delta H'_1 = 0$, thus equation (21) gives

$$\delta a' = \frac{.276 \times 7.96 \times 10.35}{30} = .76,$$

and the new diameter is

$$10.35 + .76 = 11.11 \text{ inches.}$$

The L.P. cylinder is no longer central between the main bearings, but that makes no very sensible difference in the stresses coming on the shaft.

Crank Webs.—Supposing ϵ', the fore-and-aft thickness of the webs, to remain the same, we must change ε', their breadth, to make up the strength for the increased bending moment.

K' in equation (23) now is

$$\frac{16.28}{2} + .52 + 5 + 7.65 \times 1.5 + 12.39 = 37.52,$$

and K'_1, as before, is

$$\frac{12.39}{2} + \frac{7.65}{2} = 10.02.$$

Thus from equation (23) we get

$$\frac{64.11 \times 30^2}{7.65\ \varepsilon'}\left(\frac{30}{\varepsilon'} + \frac{1.128 \times 37.52 - 1.429 \times 10.02}{7.65}\right) = 3,127.$$

By trial, or solving the quadratic equation involved and taking the positive root, we get

$$\varepsilon' = 14.00 \text{ inches.}$$

The value of ε' in *Example I* is 11.55 inches, and it is instructive to observe what a large increase has been required to compensate for the changes supposed in *Example II*; although this increase being an addition to the breadth, very greatly adds to the strength for withstanding turning moments. It brings out very strongly the importance of taking careful account of the bending moments in determining the dimensions of the shaft, and the same lesson is taught by the increase of diameter required for the main journal in this example.

It must have been a matter of surprise to many that while rules for determining the diameters of crank shafts were given by the various Boards regulating marine practice, no attempt seems to have been made to regulate the scantlings of the webs, though their fracture is perhaps as frequent as the failure of the round parts of the shaft. We also see how bad was the practice followed in several engines of the old Navy, and in some built for the merchant marine about the same time, of making broad webs which were thinned down in the fore-and-aft direction. These crank shafts nearly always gave trouble by fracturing through the webs. In wrought-iron shafts, which these were, the grain of the iron runs longitudinally across the webs, making the material weak relatively to that in the journal, and where this material is used the webs should therefore be made specially strong.

Again, in a shaft forged in one piece, frequently the material of the webs and crank pin is weaker than that in the main journals, as the former receives less working under the hammer. No allowance has been made for this in the investigation, since the metal from all parts of the shaft has to withstand the specified tests, and it would of course be unwarranted to suppose that that in the journal always exceeds this quality.

It should be noted that in one important respect the various formulas which have been framed for determining the diameters of crank shafts in the merchant marine, take no sufficient account of the bending moments. They all, I think without exception, in expressing the diameter of the main journal, contain the cube

root of the absolute steam pressure as a factor. Table I, columns I and II, show that while this would express the true law, were turning moments alone concerned, it takes no account of the increased arms for the bending moments required by the increase of F_m. It might be replied that in no case do the bending moments increase the stress in the aft journal sensibly. But we have just seen that the aft journal is not usually the part to be considered, and an inspection of Fig. VIII will show that the crank pin diameter rises a good deal more quickly than the cube root of F_m; and the same is true of the main journal diameter, especially for the dotted curve.

Finally, it only remains to be noticed that the curves in Fig. VIII can be used to determine the size of shaft which will have any desired stress. Suppose an engine is to have a shaft of 80,000 pounds steel and it is desired to raise the stress to 15,000 pounds for the bearings in line. If for the engine $F_m = 60$, it is only necessary to make the shaft of the dimensions given for $F_m = 60 \times \frac{11,500}{15,000} = 46$. For, obviously, all stresses are in proportion to the forces applied by the piston rods. (If in any particular engine these latter are doubled, all bending and turning moments, and consequently all stresses, are doubled also.) Changes in shaft dimensions for a change of b, &c., would still be determined from the formulas given above; only in equations (22) and (23) the value of F_m to be used is 46 and not 60. The pressures per square inch on the bearings would also rise in the proportion of 11,500 to 15,000. Therefore, if it was desired to run the crank pin bearing at say 500 pounds per square inch, it would be necessary to determine the alteration of bearing length, &c., which would be required to give a pressure per square inch on the crank pin of

$$\frac{500 \times 46}{60} = 383 \text{ pounds},$$

when $F_m = 46$, as already fully explained.

In what follows, this transformation will be used when determining the limit, spoken of at the beginning of the paper, at which the stresses due to steam and inertia combine to induce greater stresses in the shaft than those due to steam alone.

(*To be continued.*)

TESTS OF RIVETED JOINTS MADE AT THE WATERTOWN ARSENAL, MASS., FOR MESSRS. EDWARD KENDALL & SONS, CAMBRIDGEPORT, MASS., MAY 29, 1896.

[The following description and results of the tests made at the Watertown Arsenal were kindly supplied by the designers and makers of the joints, Messrs. Edward Kendall & Sons, Proprietors of the Charles River Iron Works, Cambridgeport, Mass.—EDITOR.]

These tests were instituted to find the relative efficiencies of two different pitches in the same style of joints. It will be noticed that the joints are made in pairs, with the difference in pitch carried out. The results seem to show that there is only a small difference in efficiency when the rivets are placed closer together. Indeed, with the smaller pitch and an excess of rivet strength over plate strength, the strength of the joint is greater. There is, however, one point which brings up a question which might well be a subject for future investigation. This is, what effect does the gripping force of the rivets have on the joint? As the rivets are driven hot, they contract in cooling and therefore do not completely fill the holes. It was noticed that a permanent set occurred in nearly every case at a pressure of about 5,000 pounds per square inch. This set probably shows the point at which the gripping force of the rivets is overcome. In tests Nos. 9,437 and 8,507, the permanent set occurred at 10,000 pounds per square inch.

Taking a boiler 72 inches in diameter, with a joint of such materials and dimensions as were used in these tests, there would

be a slight permanent set at a pressure of a little over 130 pounds per square inch. This set may occur in testing. Would it not be well then to make experiments on joints of boilers under test, and under actual conditions under steam?

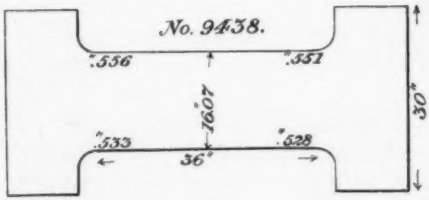
In watching the tests it is noticeable that the scale invariably starts off the rivet heads before it does on the plate. This brings up the question as to whether we do not want greater rivet strength than plate strength in the joint, and therefore more rigidity.

The efficiency of any of the complex joints is high enough to ensure the safety of a boiler during its life, but the elastic limit of the joint is below what we would expect.

It is to be hoped that other tests may be made in the direction indicated.

In the following arrangement, the cut of the plate or joint tested precedes the results obtained. For test No. 8,505, a description of the test pieces takes the place of the cut.

Steel plate representing metal used in riveted joints Nos. 9,374, 9,375, 9,376, 9,377, 9,436 and 9,437.



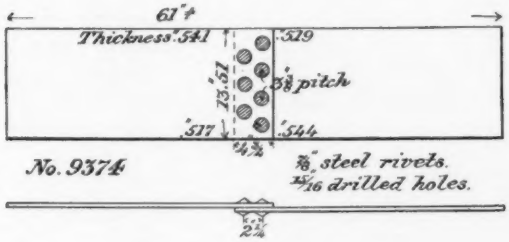
Brand on plate: "Coatsville, Pa., Fire box 55,000."

Sectional area 8.75 sq. ins. Gauged length, 30 ins. Elastic limit, 30,000 lbs. per sq. in.; elongation in inches, gauged length, .0339; set .003. At 34,000 lbs. per sq. in., the scale started off the plate.

Tensile strength, 59,680 lbs. per sq. in.

Elongation in 30 ins., 4.63 ins. = 15.4 per cent. Elongation of 5-in. sections, .68, .78, .81, .78, .79, .79 in. Minimum width of plate after the test, 15.05 in.

Plate tore apart, the fracture beginning at the neck and gradually extending across its width. Appearance silky, slightly lamellar.



Gross sectional area of plate 7.17, and net sectional area, 5.18 sq. ins. Bearing surface of rivets, 3.48, and shearing area, 4.83 sq. ins. Gauged length, 20 ins.

At 9,000 lbs. per sq. in. load, scale started off two rivet heads, the elongation being then, .0128 in. At 15,000 lbs., the elongation was .0361 in. and the set, .0175 in. At 16,000 lbs. per sq. in., scale started off plate in vicinity of rivet heads; elongation, .0459 in.

Tensile strength, 28,130 lbs. per sq. in. Sheared the rivets.

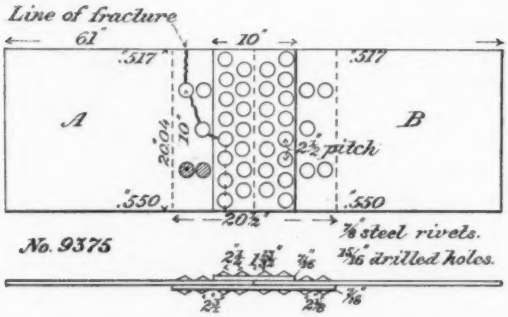
Elongation of rivet holes, outside row, .05, inside row, .02 in.

Maximum stress on joint, in lbs. per sq. in.:

Tension on gross section of plate, 28,130, tension on net section, 38,940.

Compression on bearing surface of rivets, 57,960, and shearing on rivets, 41,760.

Efficiency of joint, 47.1 per cent.



Gross sectional area of plate, 10.70, and net sectional area, 9.63 sq. ins. Bearing surface of rivets, 10.01, and shearing area, 24.15 sq. ins. Gauged length, 20 ins.

At 22,000 lbs. per sq. in., scale started off rivets, 1st and 2d rows, plate *A*; elongation, .0329 in. At 28,000 lbs., scale started off plates; elongation, .0465 in. At 30,000 lbs., the elongation was .0524 in., and the set, .0205 in.

Tensile strength, 46,930 lbs. per sq. in.

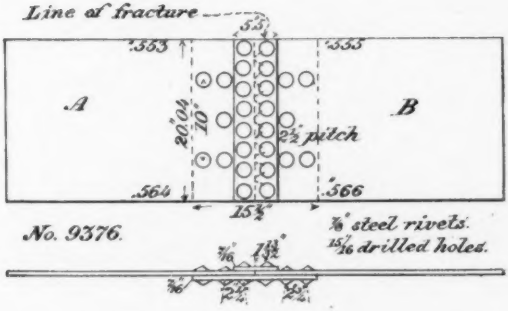
Fractured plate *A* through one rivet hole in 1st row, one in the 2d row, and four in the 3d row of rivet holes. The fracture was visible at the edge of the plate opposite the 3d row of rivets, after which fractures were in sight at the sides of the rivet holes in the 1st and 2d rows. Finally these lines of fracture extended toward each other and completed the fracture of the plate. Appearance silky, slightly lamellar. Sheared 2 rivets.

Maximum stress on joint, in lbs. per sq. in.:

Tension on gross section of plate, 46,930, and on net section, 52,150.

Compression on bearing surface of rivets, 50,170, and shearing on rivets, 20,790.

Efficiency of joint, 78.6 per cent.



Gross sectional area of plate, 11.22, and net sectional area, 10.17 sq. ins. Bearing surface of rivets, 6.82, and shearing area, 14.49 sq. ins. Gauged length, 20 ins.

At 15,000 lbs. per sq. in., scale started off rivet heads; elongation, .0219 in. At 19,000 lbs., scale generally started off the inside row of rivet heads; elongation .0291 in. At 25,000 lbs., the elongation was .044 in., and the set, .0208 in. At 30,000 lbs., scale started off plates; elongation, .06+ in.

Tensile strength, 46,530 lbs. per sq. in.

Sheared 5 rivets in plate *B*, and tore out plate in front of rivet holes at inside row.

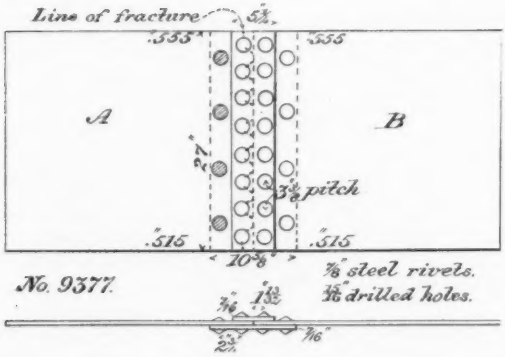
Elongation of rivet holes, 1st row .19; 2d row, end holes, .15; 3d row, middle holes, .24 in.

Maximum stress on joint in lbs. per sq. in.:

Tension on gross section of plate, 46,530, and on net section, 51,340.

Compression on bearing surface of rivets, 76,550, and shearing on rivets, 36,030.

Efficiency of joint 78.0 per cent.



Gross sectional area of plate, 14.44, and net sectional area, 12.44 sq. ins. Bearing surface of rivets, 6.02, and shearing area, 13.80 sq. ins. Gauged length, 20 ins.

At 8,000 lbs per sq. in., scale started off rivet heads, outside row; elongation, .0096 in. At 20,000 lbs., the elongation was. 04 in., and the set, .0211 in. At 29,000 lbs. per sq. in., scale started off plate; elongation, .18 in.

Tensile strength, 33,990 lbs. per sq. in.

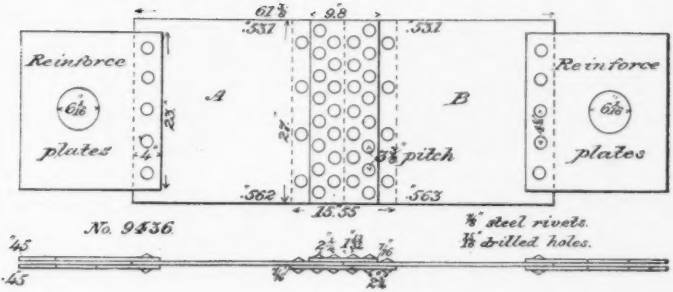
Sheared outside row of rivets in plate *A*, and tore out plate in front of rivet holes at inside row. Outside row of rivet holes elongated, .09 in.

Maximum stress on joint in lbs. per sq. in.:

Tension on gross section of plate, 33,990, and on net section, 39,450.

Compression on bearing surface of rivets, 81,530, and shearing on rivets, 35,560.

Efficiency of joint, 57.0 per cent.



Gross sectional area of plate, 14.77, and net sectional area, 12.72 sq. ins. Bearing surface of rivets, 9.74, and shearing area, 23.46 sq. ins. Gauged length, 20 ins.

At 18,000 lbs. per sq. in. there were snapping sounds, which were repeated at 19,000 lbs., and apparent again at 28,000 lbs. At 29,000 lbs. per sq. in., scale started off rivet heads; elongation, .0473 in. At 30,000 lbs., the elongation was .0524, and the set, .0241 in. At 31,000 lbs., scale started off plates; elongation, .06 in.

Maximum load reached, 47,407 lbs. per sq. in., = a total load of 700,200 lbs.

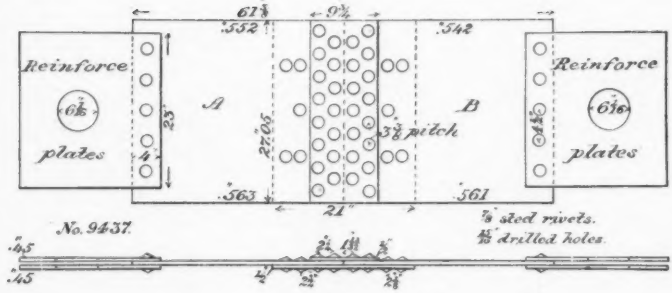
Reinforce plates buckled, allowing joint to draw from the holder jaws. Test discontinued, joint not fractured. The plates were drawn down in width to 26.61 ins., thus showing a contraction of .39 in.

Maximum stress on joint, in lbs. per sq. in.:

Tension on gross section of plate, 47,407, and on net section, 55,050.

Compression on bearing surface of rivets, 71,890, and shearing on rivets, 29,850.

Efficiency of joint, 79.4 per cent.



Gross sectional area of plate, 15.08, and net sectional area, 13.88 sq. ins. Bearing surface of rivets, 10.45, and shearing area, 24.15 sq. ins. Gauged length, 20 ins.

At 13,000 lbs. per sq. in., scale started off rivet heads in 2d row of rivets, (row of 3); elongation, .0129 in. At 28,000 lbs., scale started off plates; elongation, .0408 in. At 30,000 lbs., the elongation was .0479 in. and the set .0255 in.

Maximum load applied, 42,440 lbs. per sq. in., = a total load of 640,000 lbs.

Reinforce plates buckled, allowing joint to draw from the holder jaws. Test discontinued, joint not fractured. The plates were drawn down in width to 26.80 and 26.78 ins., thus showing contractions of .20 and .22 in. respectively.

Maximum stress on joint in lbs. per sq. in.:

Tension on gross section of plate, 42,440, and on net section, 46,110.

Compression on bearing surface of rivets, 61,240, and shearing on rivets, 26,500.

Efficiency of joint, 71.1 per cent.

The test piece, No. 8,505, of the steel plate representing metal used in riveted joints Nos. 8,506 to 8,510, was similar to that shown in cut No. 9,438. The thickness at the upper edge of the narrow part was .493, and varied at the lower edge from .498 to .500 in. The total length of the test piece was 48, and its

breadth, 24.25 ins. The narrow part was 16 ins. long and 15 ins. wide.

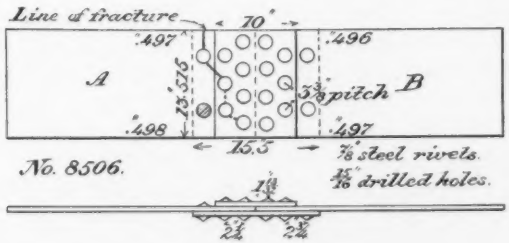
Sectional area, 7.43 sq. ins. Gauged length, 15 ins.

Elastic limit, 34,000 lbs. per sq. in., elongation, .016 in. At 35,000 lbs., the elongation was .0185 in., and the set, .0016 in.

Tensile strength, 60,550 lbs. per sq. in.; elongation, 2.25 ins.

Elongation in 15 ins., 2.43 ins., = 16.2 per cent. Elongation of inch sections: .19, .18, .16, .16, .18, .17, .17, .17, .19, .18, .16, .16, .15, .11, .10 in.

Minimum width after fracture, 13.96 ins. Fractured at the neck. Appearance silky, lamellar.



Gross sectional area of plate, 6.73, and net sectional area, 5.80 sq. ins. Bearing surface of rivets, 4.20, and shearing area, 11.04 sq. ins. Gauged length, 20 ins.

At 12,000 lbs. per sq. in., scale started off rivet heads; elongation, .0098 in. At 30,000 lbs., scale started off plate; elongation .049 in., and the set, .0286 in.

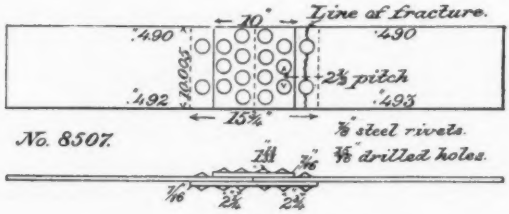
Tensile strength, 48,620 lbs. per sq. in. Sheared one rivet, and fractured plate across rivet holes in each row.

Maximum stress on joint, in lbs. per sq. in.:

Tension on gross section of plate, 48,620, and on net section, 56,410.

Compression on bearing surface of rivets, 77,900, and shearing on rivets, 29,640.

Efficiency of joint, 80.3 per cent.



Gross sectional area of plate, 4.91, and net sectional area, 3.99 sq. ins. Bearing surface of rivets, 4.14, and shearing area, 11.04 sq. ins. Gauged length, 20 ins.

At 30,000 lbs. the elongation was .0375 in., and the set, .0168 in. At 32,000 lbs. per sq. in., scale started off plate; elongation, .0495 in.

Tensile strength, 50,550 lbs. per sq. in.; elongation, .92 in.

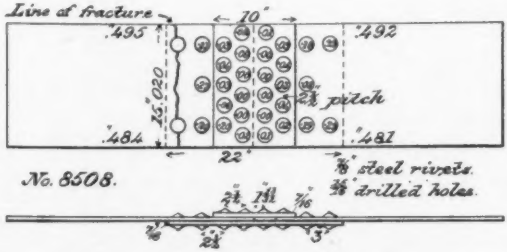
Scale did not start off rivet heads. Fractured plate across 1st row of rivet holes.

Maximum stress on joint in lbs. per sq. in.:

Tension on gross section of plate, 50,550, and on net section, 62,200.

Compression on bearing surface of rivets, 59,950, and shearing on rivets, 22,480.

Efficiency of joint 83.5 per cent.



The elongation of rivet holes after fracture is shown by the figures on the rivet heads.

Joint tested hot. Temperature of joint about 410° F.

Gross sectional area of plate, 7.35, and net sectional area, 6.43 sq. ins. Bearing surface of rivets, 7.34, and shearing area, 18.63 sq. ins.

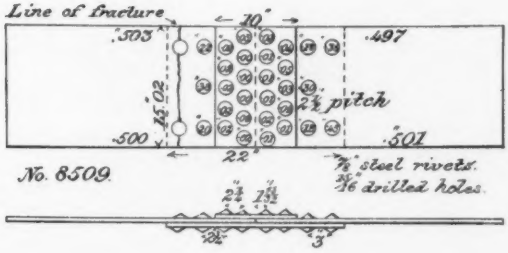
Tensile strength, 401,200 lbs., = 54,580 lbs. per sq. in.

Fractured plate across first row of rivet holes.

Maximum stress on joint, in lbs. per sq. in.:

Tension on gross section of plate, 54,580, and on net section, 62,390. Compression on bearing surface of rivets, 54,660, and shearing on rivets, 21,530.

Efficiency of joint, 90.1 per cent.



Gross sectional area of plates, 7.49, and net sectional area, 6.56 sq. ins. Bearing surface of rivets, 7.48, and shearing area, 18.63 sq. ins. Gauged length, 20 ins.

At 30,000 lbs. per sq. in., the elongation was .0395, and the set, .0191 in.

Tensile strength, 51,282 lbs. per sq. in.

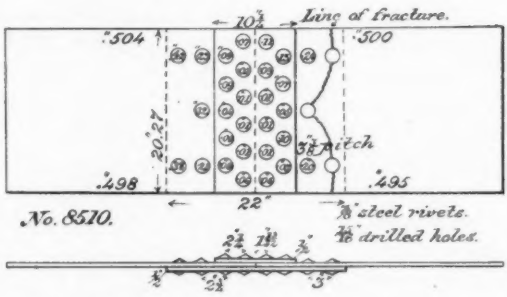
Fractured plate across first row of rivet holes.

Maximum stress on joint, in lbs. per sq. in.:

Tension on gross section of plate, 51,282, and on net section, 58,550.

Compression on bearing surface of rivets, 51,350, and shearing on rivets, 20,620.

Efficiency of joint, 84.7 per cent.



Gross sectional area of plate, 10.08, and net sectional area, 9.15 sq. ins. Bearing surface of rivets, 7.46, and shearing area, 18.63 sq. ins. Gauged length, 20 ins.

At 30,000 lbs. per sq. in., scale started off plate; elongation, .0514 in. Tensile strength, 49,960 lbs. per sq. in.

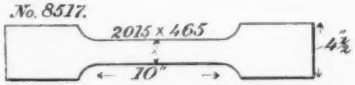
Fractured plate *B* across rivet holes in first and second row. Appearance silky.

Maximum stress on joint, in lbs. per sq. in.:

Tension on gross section of plate, 49,960, and on net section, 55,030.

Compression on bearing surface of rivets, 67,490, and shearing on rivets, 27,030.

Efficiency of joint, 82.5 per cent.



Steel plate representing metal used in riveted joints Nos. 8,515 and 8,516.

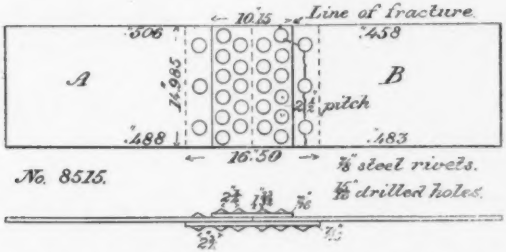
Sectional area, 2.015 ins. × .465 in. = .937 sq. in.

Elastic limit, 32,950 lbs. = 35,160 lbs. per sq. in.

Tensile strength, 56,120 lbs. = 59,890 lbs. per sq. in.

Elongation in 10 ins., 3.06 ins. = 30.6 per cent. Elongation of inch sections, .23, .27, .27, .39, .27, .25, .24, .24, .24, .21 in.

Area at fracture, 1.47 × .30 in. = .441 sq. in. Contraction of area, 52.9 per cent. Appearance of fracture, silky.



Gross sectional area of plates, 7.05, and net sectional area, 5.73 sq. ins. Bearing surface of rivets, 6.17, and shearing area, 17.25 sq. ins. Gauged length, 20 ins.

At 29,000 lbs. per sq. in., scale started off plate; elongation, .0335 in. At 30,000 lbs., the elongation was .0407, and the set, .0218 in. Tensile strength, 51,205 lbs. per sq. in.

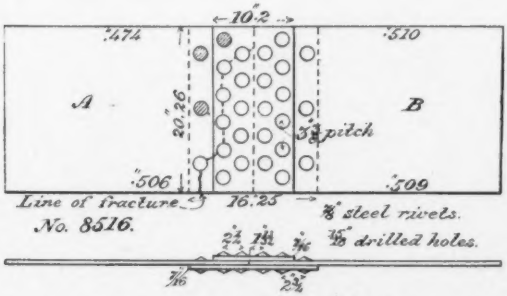
Fractured plate *B* across outside row of rivet holes, extending to the end rivet of the second row.

Maximum stress on joint, in lbs. per sq. in.:

Tension on gross section of plate, 51,205, and on net section, 63,000.

Compression on bearing surface of rivets, 58,510, and shearing on rivets, 20,930.

Efficiency of joint, 85.5 per cent.



Gross sectional area of plate, 9.93, and net sectional area, 8.55 sq. ins. Bearing surface of rivets, 6.43, and shearing area, 17.25 sq. ins. Gauged length, 20 ins.

At 25,000 lbs. per sq. in., scale started off rivet heads, outside row; elongation, .0298 in. At 26,000 lbs., scale started off plate; elongation, .0327 in. At 30,000 lbs., the elongation was .0445, and the set, .0234 in.

Tensile strength, 51,088 lbs. per sq. in.

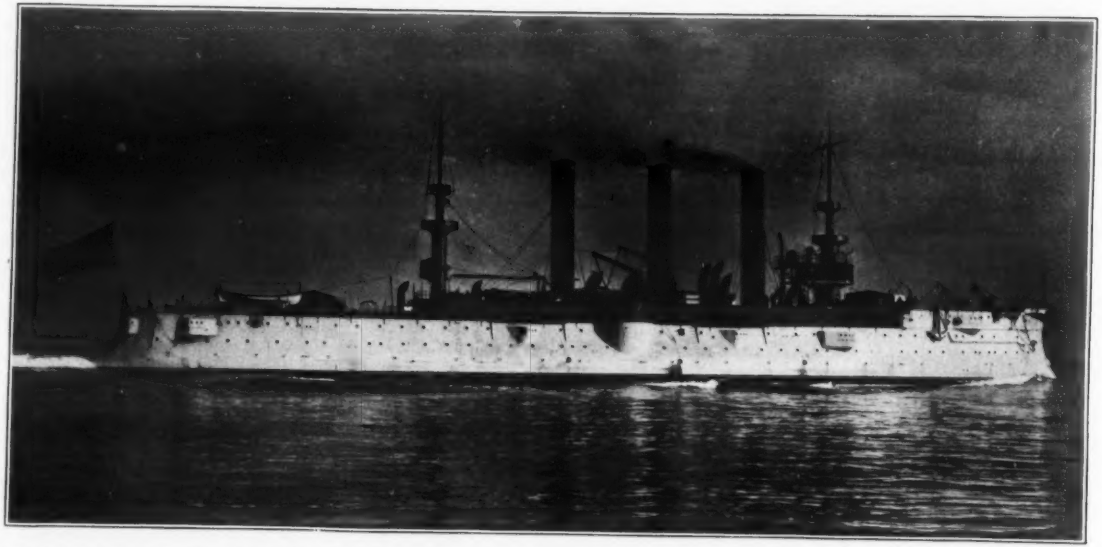
Sheared three rivets and tore plate across rivet holes in each row.

Maximum stress on joint, in lbs. per sq. in.:

Tension on gross section of plate, 51,088, and on net section, 59,330.

Compression on bearing surface of rivets, 78,900, and shearing on rivets, 29,410.

Efficiency of joint, 85.3 per cent.



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U. S. S. "BROOKLYN."

CONTRACT TRIAL OF THE U. S. ARMORED CRUISER *BROOKLYN*.

By Passed Assistant Engineer W. C. Herbert, U. S. N.

The *Brooklyn* is a twin screw armored cruiser of about 9,270 tons displacement. She was built by the Wm. Cramp and Sons' Ship and Engine Building Company of Philadelphia, Pa., from Government designs, the contract price being $2,986,000, of which it was estimated that the machinery would be $986,000, the remainder being hull and fittings. Proposals for the construction of this cruiser were issued on September 28, 1892, bids were opened on December 15, 1892, and the contract awarded on February 11, 1893. The contract requirement as to speed was that the vessel should maintain a speed of 20 knots for four consecutive hours on a displacement of 8,150 tons and with an air pressure in the fire rooms not exceeding 2½ inches, it having been further stipulated that a premium of $50,000 should be paid to the contractors for every quarter knot made in excess of this requirement, and that a penalty of $50,000 should be exacted from them for every quarter knot deficiency.

By the successful completion of the speed trials on August 27, 1896, the builders received a premium of $350,000.

HULL.

The hull is built of mild steel, which, like everything else in the vessel, is of domestic manufacture. Between frames Nos. 35 and 82, or the space occupied by the machinery, the transverse frames are spaced 4 feet between centers, and forward and abaft this space, 3.5 feet. The flat keel is made of two thicknesses, the outer plate of 25 and the inner plate of 23 pounds per square foot. The vertical keel is 42 inches deep from frames Nos. 6 to 96, tapering down forward and abaft these points; it is of 20 pounds per square foot amidships for 68 frames and of 17½ pounds for-

ward and aft of this. The outside hull plating varies from 23 pounds per square foot for one-half the length of the vessel amidships to 21.5 and 20 pounds forward and aft of this. The inner bottom is made of plates varying from 12½ to 18 pounds per square foot.

The ship is divided into 242 water tight compartments. There is a double bottom, 3 feet 6 inches deep amidships, divided into 13 water tight compartments, and extending from frame No. 22 to frame No. 82 and about 29 feet outboard. There is a cofferdam on each side, 3 feet 6 inches wide, extending the whole length of the ship between the protective and berth decks. This cofferdam is filled with cellulose at a density of 7.5 per cubic foot. The transverse and wing bunker bulkheads are carried through the cofferdams and divide them into water tight compartments.

Inboard of the cofferdams there is a passageway on each side, two feet wide and extending from frames Nos. 35 to 82. Below the protective deck, there is a wider passageway on each side of the engine and boiler compartments. Both sets of passageways are divided into smaller compartments by the transverse water tight bulkheads, and thus form practically an extension of the double bottom. At each extremity of the vessel, several compartments are used for trimming tanks.

Hold.—In the hold abaft the engine compartments and forward of the boiler compartments are 5-inch and 8-inch magazines and shell rooms, fixed ammunition rooms, store rooms and trimming tanks.

After Platform.—On the after platform are fixed and small arms ammunition and handling rooms, air compressor, blower room, store rooms, steering engine and gear.

Middle Platform.—On the middle platform, between frames Nos. 53 and 57, are the 8-inch turret turning engines, 8-inch shell rooms, 5-inch magazines and ammunition hoist.

Forward Platform.—On the forward platform are coal bunkers, dynamo rooms, ammunition handling room, air compressor, store rooms and trimming tank.

Orlop Deck.—On the orlop deck, beginning forward, are store rooms, anchor gear, and fresh water tanks; coal bunkers, blower

and distiller rooms, 8-inch turret turning machinery, the passage-ways and cofferdams, and aft more store rooms.

Berth Deck.—On the berth deck are the sick bay and dispensary forward, the living quarters, the engineers' workshop, distiller room, and four torpedo tubes.

For pumping and drainage there is a 14-inch main drain on the port side and a $7\frac{1}{2}$-inch secondary drain on the starboard side. Sluice valves, worked from the deck, allow the passage of water from one compartment to the other. The auxiliary pumps having bilge suctions draw from the secondary drain pipe. Valve boxes in the forward fire room and after engine room give pump connections to all the compartments and trimming tanks.

There are 36 coal bunkers above the protective deck holding 735 tons, and 19 bunkers below the protective deck holding altogether 958 tons. Total bunker capacity at 43 cubic feet per ton, 1,693 tons.

In order to give more fore-and-aft fire to the 8-inch guns in the midship turrets, the sides of the *Brooklyn* tumble home, as shown in the sectional views.

The following boats are carried:

1 33-foot steam cutter.
1 30-foot steam cutter.
2 30-foot whale boats.
1 30-foot whale boat gig.
1 33-foot sailing launch.
1 30-foot barge.
4 28-foot sailing cutters.
2 20-foot dinghies.
2 Balsas.

Length between perpendiculars, feet and inches	400–6
on L.W.L., feet and inches	400–6
over all (including rudder), feet and inches	402–$7\frac{1}{2}$
Beam, extreme, and at L.W.L., feet and inches	64–$8\frac{1}{4}$
Ratio of length to beam	6 to 1
Depth in hold from top of main deck beams to top of floor, feet and inches,	33–$10\frac{3}{16}$
Draught, forward and aft, seagoing trim, feet	24
Displacement, seagoing trim (load draught), tons	9,271
per inch, at L.W.L., tons	41.19

Area of immersed midship section, square feet	1,367
L.W.L. plane, square feet	17,433
Center of gravity of L.W.L. plane, aft of midship section, feet and inches,	4–2½
buoyancy above bottom of keel, feet and inches	13–11½
aft of midship section, feet and inches	1–2½
gravity above bottom of keel, feet and inches	24–4½
Transverse metacenter above center of buoyancy, feet and inches	13–4½
Longitudinal metacenter above center of buoyancy, feet	425
Coefficient of fineness on extreme dimensions	.520
of midship section	.881
of L.W.L.	.673
Cylindrical coefficient	.586
Number of frames	106
water tight compartments	242

ARMOR.

The protective deck armor over the machinery space consists of two courses of 1½-inch steel plates. Forward and aft of this, the total thickness of the two plates is not less than 2½ inches. The protective deck extends the whole length of the ship. Glacis plates, 3 inches thick, are fitted around the engine hatch.

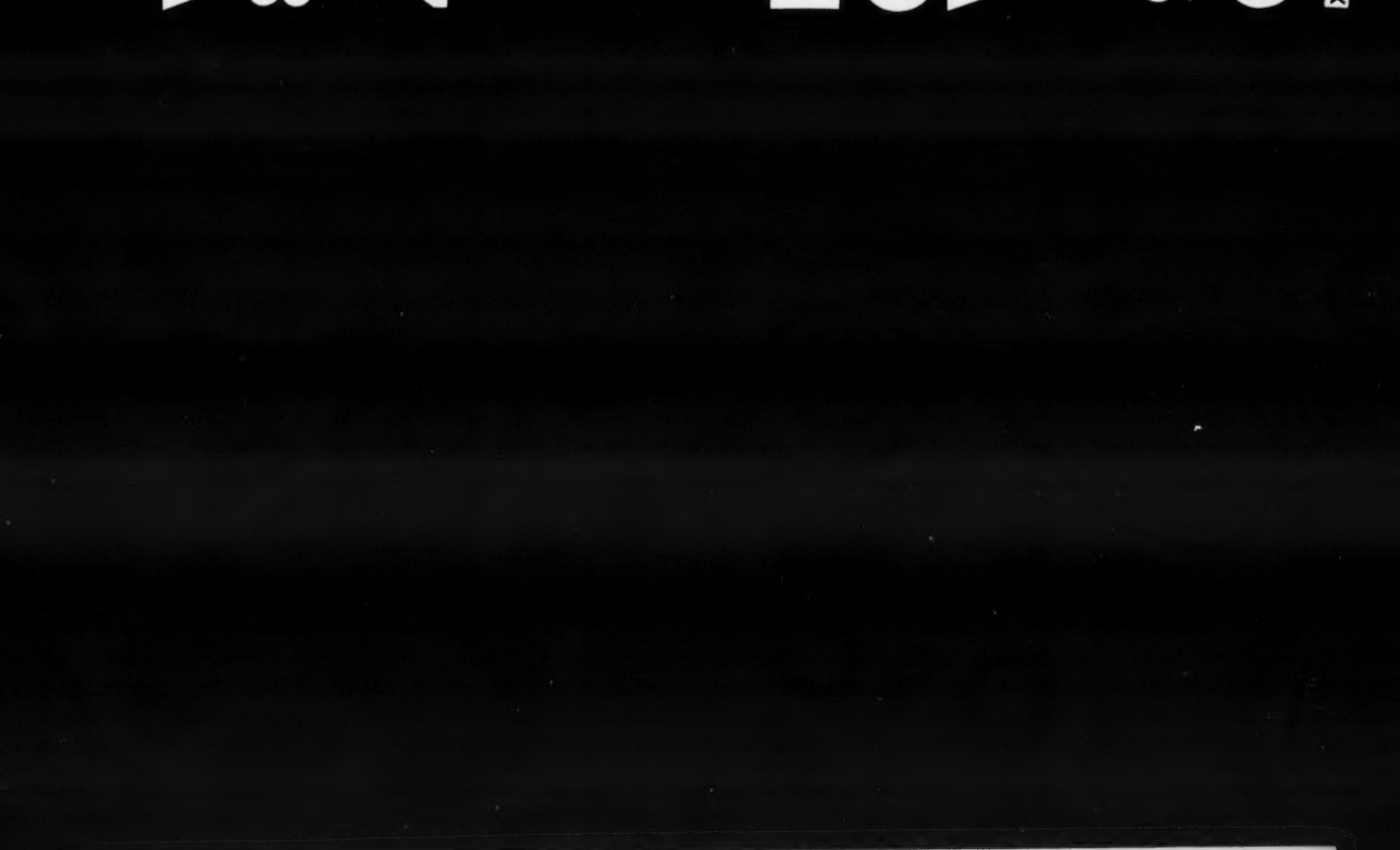
The side armor is 3 inches thick, and extends from 4 feet above to 4 feet below the 24-foot water line for a length of about 192 feet opposite the engine and boiler spaces. The barbette armor of 8-inch turrets is 8 inches thick, with a thickness of 4 inches where not exposed. The turret armor is 5½ inches thick, secured to a backing plate of 20 pounds per square foot. The side armor, turret armor and barbettes are of harveyized nickel steel.

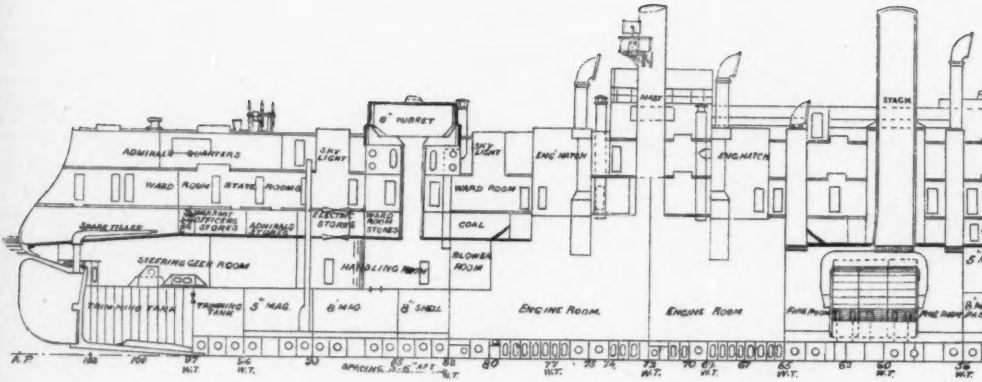
The armor of the 5-inch gun sponsons is 4 inches thick and the splinter bulkheads, 1½ inches thick. The secondary battery protection is 2 inches thick.

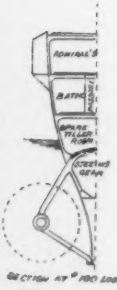
The conning tower and shield are of forged steel, 7½ inches thick; from the center of the conning tower an armor tube, 5 inches thick and 12 inches in internal diameter, runs down into the forward handling room.

ARMAMENT.

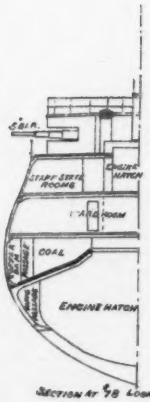
There are eight 8-inch guns mounted in pairs in four turrets. The guns in the forward and after turrets have an angle of fire of 290°, or 145° on each side of the bow and stern.



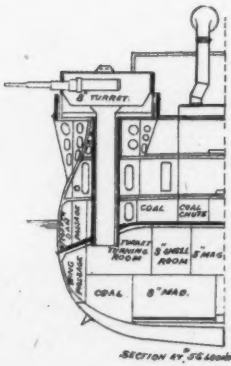




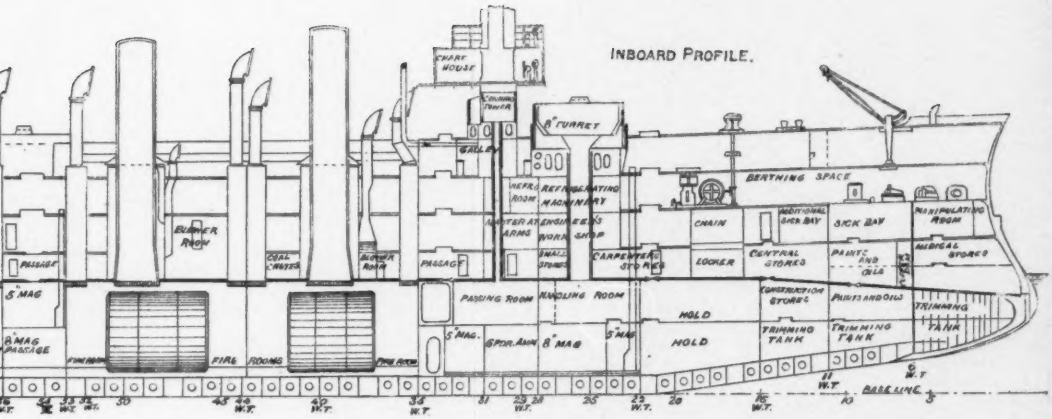
SECTION AT 97 FOR LOOKING FORD.

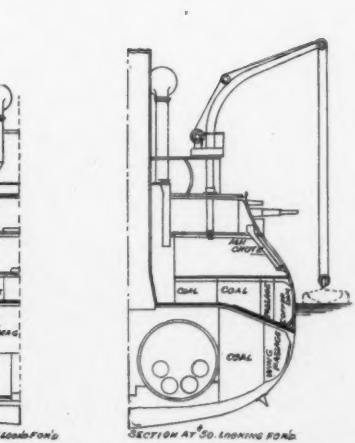


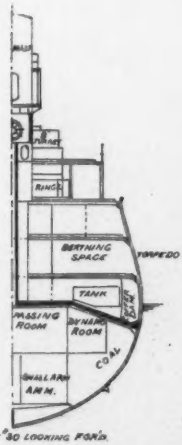
SECTION AT 73 LOOKING FORD.



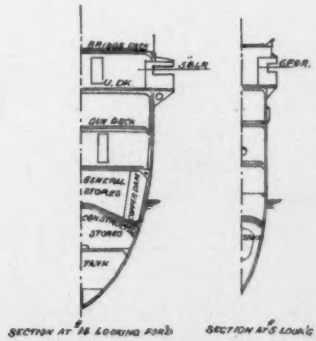
SECTION AT 55 LOOKING

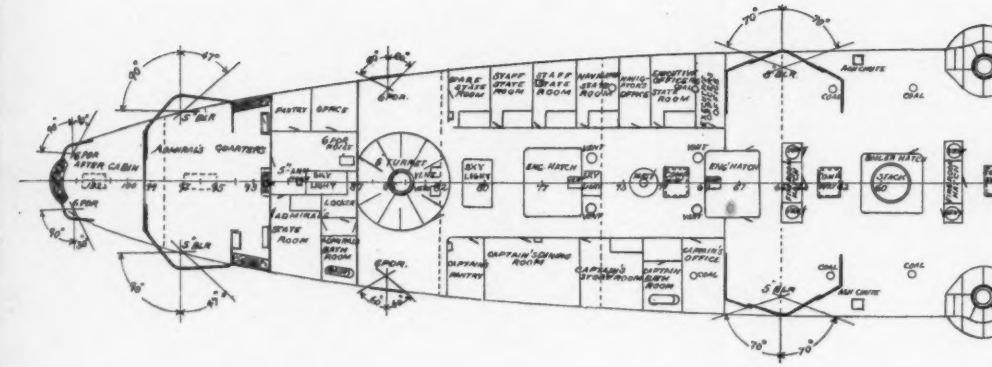

INBOARD PROFILE.
CHART HOUSE
8" TURRET
BERTHING SPACE
REFRIGERATING MACHINERY
ENGINEERS WORK SHOP
CHAIN
LOCKER
CENTRAL STORES
SICK BAY
MEDICAL STORES
CARPENTER'S STORES
PASSAGE
5" MAG
8" MAG
PASSING ROOM
HANDLING ROOM
HOLD
TRIMMING TANK
FIRE ROOM
BASE LINE

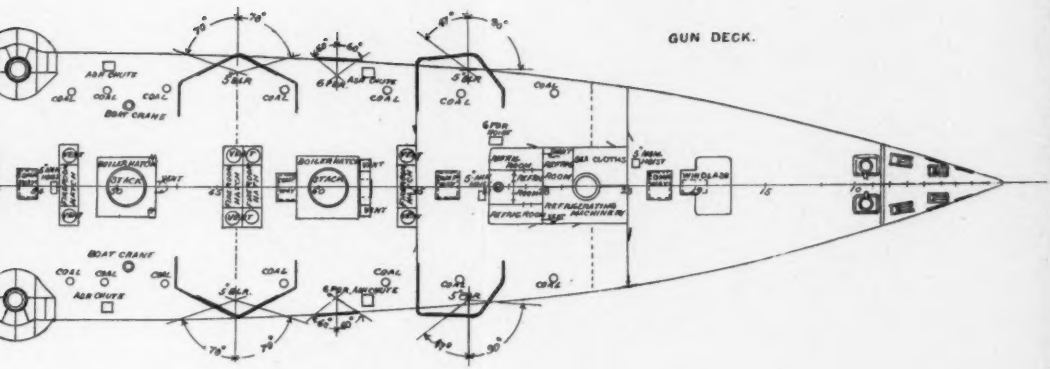

SECTION AT 50. LOOKING FOR'D.

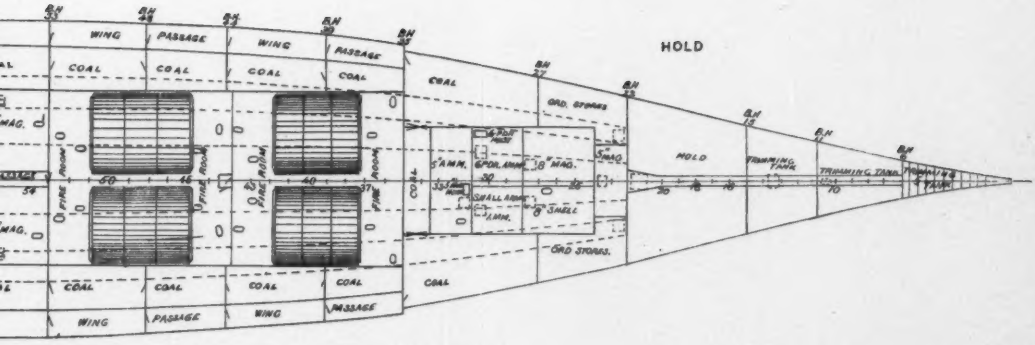

SECTION AT 80 LOOKING FOR'D.

SECTIONS.


SECTION AT 18 LOOKING FOR'D
SECTION AT 5 LOOK'G FOR'D


ADMIRAL'S QUARTERS
PANTRY
OFFICE
STATE ROOM
ADMIRAL
BATH ROOM
LOCKER
PASSAGE
ENG. HATCH
ENGINE ROOM
BOILER HATCH
STACK
COAL
CAPTAIN'S DINING ROOM
CAPTAIN'S STATE ROOM
CAPTAIN'S OFFICE
CAPTAIN'S BATH ROOM
CAPTAIN'S PANTRY
ENGINEER'S OFFICE
ENGINEER'S STORES
WASTE STORES
ENGR'S STORES
WING PASSAGE
TRIM'S TANK
8" MAG.
8" SHAFT
8" SWELL


GUN DECK.
ASH CHUTE
COAL
BOAT CRANE
5" GUN
6 POR. ASH CHUTE
BOILER HATCH
STACK
VENT
6 POR. HATCH
REFRIGERATING MACHINERY
REFRIG. ROOM
GUN CLOTHS
3" MAN. HOIST
WIRE CLOTH
6 POR. ROOM


HOLD
WING
PASSAGE
COAL
GRD. STORES
FIRE ROOM
TRIMMING TANK
6 POR. AMM.
3" MAG.
SMALL ARMS AMM.
3" SHELL
6 POR. ROOM

The midship guns have an angle of fire of 180° from right ahead to right astern. There are twelve 5-inch rapid fire guns mounted in sponsons on the gun deck. There are, in addition, twelve 6-pounders, four 1-pounders and four machine guns. The 1-pounders and machine guns are mounted on the rails and in the tops.

The ammunition is supplied by electric hoists, ten in number. There are four above-water torpedo tubes, two on each side, for firing Whitehead torpedoes. The air compressors are of the Rand Drill Co.'s three-stage type, and, together with the accumulators, are situated in the forward and after handling rooms.

MAIN ENGINES.

There are four vertical, direct acting, three cylinder, triple expansion engines, placed one in each of four separate water tight compartments, connected by five water tight doors, one on each side in the athwartship bulkhead, one in the fore-and-aft bulkhead between the forward compartments, and two in the fore-and-aft bulkhead between the after compartments. All these doors are on a level with the working platform. The starting gear is inboard.

There are two engines on each shaft and, in place of the disconnecting coupling fitted on the *New York*, there are four taper coupling bolts, the coupling being of the ordinary disc kind.

The I.P. and L.P. cylinders are fitted with steam jackets, the space left around the working linings being $\frac{3}{4}$ inch. The H.P. cylinder liners are secured at the top by countersunk steel bolts. The I.P. and L.P. liners are left free to expand at the top, the joint being made by a packing ring and packing. The valves are all of the single ported, piston type, made of cast iron, one for the high pressure and two each for the intermediate and low pressure cylinders. The low pressure valves are balanced by making the upper ends $1\frac{1}{2}$ inches larger in diameter than the lower, the live steam being between the ends. The other valves are fitted with balancing pistons, connected above with the condensers. The valve gear is of the double bar Stephenson

link type. The cut-off can be varied from about .5 to .7 of the stroke by means of a slot in the reversing arm.

Reversing is effected by a Cramps' steam reversing gear, which consists of a steam lift secured to one of the engine frames, and connected to an arm on the reversing shaft. The lifting piston is operated by a piston valve which is controlled by a floating lever, receiving motion at one end from the hand lever, and a reverse motion at the other from a pin on the crosshead, so that the piston moves and stops with the hand. The diameter of the reversing cylinders is 14 inches, and the stroke 19 inches. The reversing gear of both engines on one shaft can be operated by one lever in either engine room, or at the door between the engine rooms; or either forward or after hand lever may be disconnected from the others. Each engine has a detachable hand reversing gear, which consists of a hand wheel, worm shaft, pinion and rack attached to a separate arm of the reversing shaft.

Each engine is fitted with a disc stop valve, having a screw stem and a balancing piston, and a butterfly throttle. The former is $12\frac{3}{8}$, and the latter, 14 inches in diameter. The main pistons are of cast steel, dished, and fitted with two packing rings, each $\frac{5}{8}$ inch wide and $\frac{3}{4}$ inch deep, and the followers are grooved. The crank and wrist pins are fitted with slab brasses, the former brasses as well as the eccentric straps being lined with Parson's white metal. The eccentric straps are made of composition. The crossheads are of forged steel, bolted to cast steel shoes, faced with Parson's white metal. The crosshead pins are 7 inches in diameter and 8 inches long, with a $2\frac{1}{2}$-inch hole.

The engine frames are of the inverted **Y**-type, of cast steel, two for each cylinder; each frame is made in two sections which are bolted together in the vertical plane. Cast on the inside of each frame are ribs and facings to which the cast iron crosshead guides are bolted. The space between the frame and the guide is used for water circulation. The bed plates are of cast steel of **I**-section, each in three sections, bolted together. The bed plates of forward engines are fitted like a pillow block and its wedge,

so that the forward shaft may be adjusted to the after engine shaft whenever the bearings wear down.

The exhaust passages between the steam chests and to the condensers are all formed by external composition pipes. There is a copper expansion joint next to the condenser in each main exhaust pipe. The main steam pipes, which are made of copper, are reinforced with $\frac{1}{8}$-inch steel bands at intervals of six inches. An oil pump is placed in the port after engine room, discharging to a distributing oil tank in each forward engine room. From this tank, pipes are led to the different parts of the engines.

The main engine stop valves and the main boiler stop valves and safety valves can be operated from the berth deck.

Cylinders, number of, each engine		3
diameter of H.P., starboard, forward, inches, $31\frac{1}{16}$	all others..	$31\frac{3}{8}\frac{1}{2}$
I.P., port, forward, inches, $46\frac{3}{16}\frac{1}{2}$	all others..	$46\frac{1}{16}\frac{3}{2}$
L.P., starboard, forward, inches, $71\frac{1}{16}$	aft..	72
L.P., port, forward, inches, $71\frac{3}{16}\frac{1}{2}$	aft..	$71\frac{1}{16}\frac{3}{2}$
Stroke of pistons, inches		42
Valves, diameter of H.P. (1) and I.P. (2), inches		16
L.P. (2), inches, upper, 30	lower..	28.5
Balance pistons, diameter of H.P., inches		4
I.P., inches		4.75
Area of main steam pipe, each engine (12 inches), square inches		113.10
exhaust pipe to I.P. cylinder, each engine (14.5 inches), square inches,		165.13
L.P. cylinder, each engine (20 inches), square inches,		314.16
condenser, each engine (24.5 inches), square inches,		471.43
Volume swept by H.P. piston (mean), per stroke, cubic feet		18.53
I.P. piston (mean), per stroke, cubic feet		40.36
L.P. piston (mean), per stroke, cubic feet		98.91
Clearance of H.P. cylinder, per cent	Top	23.25
	Bottom	20.17
I.P. cylinder, per cent	Top	22.41
	Bottom	21.53
L.P. cylinder, per cent	Top	19.02
	Bottom	17.96
Net area of I.P. ÷ that of H.P. piston		2.09
L.P. ÷ that of I.P. piston		2.47
L.P. ÷ that of H.P. piston		5.16
Piston rods, diameter, inches		$6\frac{3}{4}$
Connecting rods, length between centers, inches		84
diameters and thickness, inches		$6\frac{1}{4}$ and 8 x $6\frac{1}{4}$

Shafting and Bearings.—The crank, thrust and forward sections of the propeller shafts are of forged steel; the after sections of the propeller shafts are of nickel steel. The crank shaft of each engine is in three sections, the cranks being bolted to each other at angles of 120°, and the sequence for ahead motion being H.P., I.P. and L.P. The two engines on each shaft are coupled with the H.P. cranks opposite each other. The after couplings of the L.P. crank shafts are of the same dimensions as those of the after engines. The couplings are fitted with tapered, headless bolts, and split pins over the nuts. There is no shaft alley, the thrust shaft coupling direct to the after section of crank shaft of the after engine.

The propeller shafts are in two sections. The forward section is covered between couplings by a composition casing; the after section is covered where it passes through the strut bearing. The two sections are connected to each other by plain disc couplings. The coupling between the propeller and thrust shafts is as follows: The starboard or forward end of the propeller shaft is fitted with a forged steel sleeve, 15 inches long, secured to it by three steel keys. Aft of this coupling, there is a groove, six inches wide and one inch deep, cut in the shaft. In this groove a collar, made in halves, is fitted, its outside diameter being the same as that of the coupling. There is a coupling disc forged on the after end of the thrust shaft. This disc, the collar, and the keyed sleeve are secured to each other by nine $3\frac{1}{2}$-inch steel through bolts.

Where the propeller shafts pass through the couplings forward and the propellers aft, the 11-inch hole is reduced to 4 inches in diameter.

Each thrust bearing pedestal is of cast iron of the horse shoe pattern. The end and side walls of the pedestal form an oil trough inside of which, both forward and abaft the horse shoes, is a composition bearing lined with white metal for taking the weight of the shaft. The cap of this bearing is of cast iron, lined with Parson's white metal. The collars are made of a mixture of cast iron and steel, lined with white metal, and can be adjusted by brass nuts on steel side rods, the latter being $3\frac{3}{4}$ inches in

diameter. The pedestal is bolted to a cast-iron sole plate fitted with wrought-iron wedges at each end of the pedestal for adjusting the bearing fore and aft.

The crank shaft bearings have only bottom brasses, the cast-steel caps, as well as the lower brasses, being lined with Parson's white metal. Water circulation is provided through crank shaft bearings, crosshead guides and under the thrust bearing.

Crank shafts, outside and inside diameters, forward engines, inches....	13½ and 6
after engines, inches.....	17 and 7
coupling discs, outside diameter, forward engines, inches,	25½
after engines, inches.....	30½
thickness, forward engines, inches.....	3
after engines, inches.....	4
coupling bolts, taper 1 inch in 1 foot, smallest diameter, inches.....	2⅞
journals, outside diameter, forward engines, inches.....	13½
after engines, inches.....	15
length of each section, forward engines, inches.....	86, 112, 131
after engines, inches.....	100, 112, 86
Crank pins, outside and inside diameters, forward engines, inches.....	13½ and 6
after engines, inches.....	17 and 7½
length, forward engines, inches.....	17
after engines, inches.....	14¼
Crank webs, width, forward engines, inches.....	15
after engines, inches.....	18½
thickness, forward engines, inches.....	9
after engines, inches.....	11½
Thrust shafts, diameter, inches.....	16¾
diameter and thickness of forward coupling disc, inches,	30½ and 4
axial hole, inches..	7½
collars, number each shaft.....	12
diameter, inches.....	21¾
thickness, inches.....	2
distance between, inches.....	3
surface, total for both engines, square inches.....	3,628.54
length, feet.....	12.25
Propeller shafts, outside and inside diameters, forward sections, inches,	18 and 11
after sections, inches.....	17 and 11
diameter of coupling discs, inches	30½
length, forward sections, about, feet and inches.....	33–0
after sections, about, feet and inches	38–10½
Stern bracket bearing, length, inches.....	34
diameter, inches.....	25
Stern tube bearing, length, forward, inches.....	25
aft, inches.....	30¾

Main Condensers.—The main condensers, one for each engine, are made of cast brass, each in five sections, including the water chests. The water circulates through tubes. Brass baffle plates are fitted to direct the steam over the tubes, and plates are provided for supporting the tubes and also to act as baffle plates. The tube sheets are made of rolled brass, 1 inch thick. The tubes are packed with cotton tape set up by screw glands, and are spaced $1\frac{5}{16}$-inch between centers.

Length over all, feet and inches	10–9
Diameter of shell, inside, feet and inches	5–9
Thickness of shell, inch	$\frac{7}{16}$
Tubes, exposed length, feet	9
outside diameter, inch	$\frac{5}{8}$
thickness, B.W.G. No. 20, inch	.035
number, one condenser	3,684
Cooling surface, one condenser, square feet	5,425
Ratio of total cooling surface to total heating surface	1 to 1.54

Main Air Pumps.—There is one double, vertical, single acting Blake air pump for each main engine, fitted with the Blake valve gear. Both pumps on one side are connected to both condensers on the same side with intervening straightway valves, and exhaust into either the condensers or the I.P. or L.P. valve chests. The pumps are connected by a beam pivoted at its center. The beam receives its motion through links swinging from a crosshead on the pump and piston rods at each end. The pumps deliver into the feed tanks.

Diameter of steam cylinders (2), inches	12
water cylinders (2), inches	25
Stroke, inches	18
Diameter of piston rods, inches	$2\frac{1}{4}$
pump rods, inches	4
Number of foot valves (vulcanized rubber), each pump cylinder	10
bucket and delivery valves, each	8
Diameter of foot, bucket and delivery valves, inches	$5\frac{1}{2}$
Area of valves, each, square inches	13
Ratio of volume swept by L.P. piston, per stroke, to that of air pump buckets, per stroke	9.64
Ratio of volume swept by one L.P. piston (mean) to that of the two buckets of one air pump per minute, on full power trial, starboard, 128.47.....port..	149.06

These air pumps are of the same style as fitted on the *New York*, *Columbia*, *Minneapolis* and other ships, and their successful working has been described in previous reports of trials.

Main Circulating Pumps.—There is a centrifugal, double inlet circulating pump for each condenser, arranged to draw either from the sea, from the bilge of its engine room, or from the main drainage pipe. The sea and bilge injection valves are fitted with a safety lock, so that both cannot be open at the same time. The circulating pumps are driven by horizontal, simple, single cylinder, direct acting engines.

Diameter of steam cylinders, inches	12
Stroke, inches	9
Diameter of runner, inches	42
Diameter of suction, inches	14
Diameter of discharge, inches	14

Auxiliary Condensers.—There is in each after engine room a Wheeler condenser connected with the auxiliary exhaust pipes, and of sufficient capacity for one-half of all the auxiliary machinery. Each has a horizontal combined air and circulating pump, the steam cylinder being between the water cylinders, and all the pistons being on one rod. The tubes are arranged and packed as in the main condensers.

Cooling surface, one condenser, square feet	800
Diameter of steam cylinder, inches	10
water cylinder (circulating), inches	12
air pump cylinder, inches	12
Stroke, common, inches	12

Screw Propellers.—The propellers are of manganese bronze, and are three-bladed, true screws. The blades are bent back and are adjustable from a pitch of 19 feet 6 inches to 22 feet 3 inches. The hubs are spherical and fitted with conical tail pieces. The starboard propeller is right, and the port one left handed.

Number of blades	3
Diameter of propeller, feet	16.5
Diameter of hub, inches	54
Length of hub, inches	43
Pitch, as set on trial, starboard, feet, 19.98 ..... port..	20.01
Helicoidal area for one screw, square feet	75.74

Disc area for one screw, square feet	231 82
Pitch ÷ diameter	1.21
Disc area of both screws ÷ I.M.S.	.311
Immersion of center at mean trial draught, feet and inches	13-7½
Center above lowest point of keel, feet	8.5

Speed Indicators.—In each after engine room there is a mechanical device for indicating the speed of both main shafts. Each indicator is fitted with two indices pivoted at the center of a graduated disc, and connected by shafting and gearing, one to the starboard, and the other to the port main shaft. The pointers are so geared that they have a much slower speed than the main shafts, and the graduations on the disc are numbered for revolutions of the main shafts, so that the number of graduations traversed by the indices per minute indicates the speed of the engines. The indices are also so arranged that they can be set together at any time and the relative speeds of the two engines ascertained. It is proposed to fit electro-dynamic speed indicators to register in the conning tower and chart house. These indicators have shown very successful tests, a movement of the pointer having been shown when the engines were turned by the turning engines.

BOILERS.

Steam is supplied from five double ended and two single ended steel boilers, all 16 feet 3 inches in diameter. Four of the double ended boilers are 18 feet, and the fifth, 19 feet 11½ inches in length. The single ended boilers are both 9 feet 5 inches long.

The working pressure of all boilers is 160 pounds per square inch. There are four Fox's corrugated furnaces in each end of each double ended boiler and four in each single ended boiler.

The boilers are all below the protective deck and placed in three water tight compartments, separated by two athwartship bulkheads. Two double ended boilers are placed in the forward, and two in the after compartment. In the middle compartment, the larger double ended boiler is on the port side, and the two single ended, placed back to back, are on the starboard side.

There is one smoke pipe for the boilers in each compartment. These pipes are square at the base and pass into a circular section about 9 feet above the protective deck. The part passing

through the protective deck is $\frac{1}{2}$ inch thick. The usual armor bars are fitted inside. The upper parts of the pipes are stayed to each other and to the ship by two rows of fore-and-aft guys and athwartship guys fitted with turnbuckles. The principal feature of these pipes is their height, the top of each pipe being 100 feet above the level of the grate bars. The *Brooklyn* is the first ship of our Navy to be fitted with the tall smoke pipes, and the results of the trial trip show that the increase in the height of the smoke pipes relieved matters in the fire room very much, the necessary steam pressure being maintained without running the air pressure up to the $2\frac{1}{2}$-inch limit.

The longitudinal shell seams of the boilers are treble riveted with double butt straps. Joints of boiler heads and shell seams are double riveted, and the other circumferential seams are lapped and treble riveted. The front and back heads of all boilers are curved at the top; the radius for the double ended boilers being 3 feet 10 inches, and for the single ended boilers, 3 feet 2 inches.

The boiler tubes are of charcoal iron, lapwelded and drawn. The ordinary tubes are swelled to $2\frac{5}{16}$ inches external diameter at the front ends. The back ends are expanded in the tube sheets and beaded over. The stay tubes are reinforced at both ends to an external diameter of $2\frac{3}{8}$ inches. They are then swelled to $2\frac{1}{2}$ inches at the front ends. They are screwed into both sheets, the front ends being made tight by the taper screw, and the back ends, by expanding and beading. Cast-iron ferrules, $1\frac{1}{2}$ inches in internal diameter, are used to protect the ends of the stay tubes in the combustion chambers.

The furnaces are fitted with Cone's patent cast iron shaking grates. There are 8 grate bars in each furnace of the main boilers, each bar extending the whole length of the furnace. They rock on lugs on the front and back bearers and on projections on the middle bearers, and can be easily renewed without hauling fires.

The boilers are fitted with a steam circulating apparatus, and the internal feed pipes are arranged to distribute the feed water throughout the boilers.

DOUBLE ENDED BOILERS.

Length, feet and inches, four of 18 and one of	19–11½
Diameter, outside, feet	16.25
Thickness of shell and top heads, inches	1 $\frac{23}{64}$
heads, bottom, inch	$\frac{3}{4}$
tube sheets, inch	$\frac{7}{8}$
furnaces, inch	$\frac{9}{16}$
Combustion chambers, 4 in each boiler, thickness, inch	$\frac{9}{16}$
depth at top, inches	23
width, inches	84
Furnaces, greatest internal diameter, feet and inches	3–8
least internal diameter, feet and inches	3–4
length of grate, feet and inches, smaller boilers, 6–4, larger,	6–6
number in each boiler, corrugated	8
Tubes, outside diameter, inches	2¼
length between tube sheets, feet and inches, smaller boilers, 6–6, larger	7–5¾
number of ordinary (B.W.G. No. 12)	904
stay (B.W.G. No. 6)	296
spaced vertically, inches	3½
horizontally, inches	3½
Diameter of rivets in shell sheets, inches	1 $\frac{5}{16}$
screw stays between backs of combustion chambers, inches,	1 $\frac{7}{16}$
Number and diameter of through upper braces, inches	10 of 2¾
lower braces, inches	2 of 2
braces around each lower manhole, inches	3 of 1¾
from head to back tube sheets, inches	2 of 2½, 6 of 2
Heating surface, tube, square feet, smaller boilers, 4,594.....larger..	5,286
furnace and combustion chambers, square feet, smaller boilers, 842.....larger..	890
total square feet, smaller boilers, 5,436.....larger..	6,176
Grate surface, square feet, smaller boilers, 168.6.....larger..	173.2
Area through tubes, square feet	25.85
over bridge walls, square feet	19.67
Smoke pipes (3), diameter, feet	7.25
area of all, square feet	123.84
height above lowest grates, feet	100
Diameter of boiler main stop valves, inches	10
auxiliary stop valves, inches	7

SINGLE ENDED BOILERS.

Length, feet and inches	9–5
Diameter, outside, feet	16.25
Thickness of shell and top heads, inch	1 $\frac{23}{64}$
heads, bottom, inch	$\frac{3}{4}$
tube sheets, inch	$\frac{7}{8}$

Thickness of furnaces, inch	$\frac{9}{16}$
Combustion chambers, each boiler	2
thickness, inch	$\frac{9}{16}$
depth at top, inches	23
width at top, inches	$85\frac{1}{2}$
Furnaces, greatest internal diameter, inches	$42\frac{7}{8}$
least internal diameter, inches	40
length of grate, feet and inches	6–4
number in each boiler (corrugated)	4
Tubes, outside diameter, inches	$2\frac{1}{2}$
length between tube sheets, feet and inches	6–$4\frac{3}{4}$
number of ordinary (B.W.G. No. 12)	466
stay (B.W.G. No. 6)	152
spaced vertically, inches	$3\frac{1}{4}$
horizontally, inches	$3\frac{1}{2}$
Diameter of rivets in shell sheets, inches	$1\frac{1}{16}$
screw stays, inches	$1\frac{3}{8}$
Pitch of screw stays, inches, horizontally, $7\frac{3}{4}$ ..... vertically	7
Number and diameter of through upper braces, inches	10 of $2\frac{3}{4}$
lower braces, inches	3 of $1\frac{3}{4}$
braces from head to back tube sheets, inches,	6 of 2
Heating surface, tube, square feet	2,335
furnaces and combustion chambers, square feet	421
total, square feet	2,756
Grate surface, square feet	84.3
Area through tubes, square feet	13.3
over bridge walls, about, square feet	6.4
Number and diameter of safety valves (2 on one base), inches	2 of 5
Diameter of boiler stop valves, inches	$7\frac{1}{2}$

TOTALS FOR ALL BOILERS.

Heating surface, tube, square feet	28,332
furnaces and combustion chambers, square feet	5,100
total, square feet	33,432
Grate surface, square feet	1,016.2
Area through tubes, square feet	155.86
Ratio, total H.S. to G.S.	32.9 to 1
area through tubes to G.S.	.153 to 1
total area of smoke pipes	1.26 to 1

Forced Draft.—The closed fire room system is used, there being in each fire room two Sturtevant blowers, each driven by a double engine. The diameter of the steam clyinders is 5 inches and the stroke is 4 inches. The diameter of the fan is 60 inches and its width, 18 inches.

Feed Pumps.—There are three main and four auxiliary Blake feed pumps in the fire rooms. Both the main and auxiliary feed pumps are so connected that any pump will supply any boiler, but there is no connection between the main and auxiliary systems. There are also independent connections with the feed tanks. Two of the engine room pumps are connected with the feed tanks and main feed systems. These pumps are also arranged to draw from the channel pipes of the main condensers. The main feed pumps draw from the feed tanks only, and deliver to the boilers only. The auxiliary feed pumps draw from the tanks, the bilge, the secondary drains, the boilers and the sea, and deliver into the boilers, the fire main and overboard.

The following table gives the number and sizes of main and auxiliary feed pumps. The dimensions of the cylinders and the stroke are in inches and the capacity in gallons:—

No.	Purpose.	Steam cylinder.	Water cylinder.	Stroke.	Capacity.
1	Main feed	12	7⅜	12	350
2	Main feed	10	6⅝	12	250
1	Auxiliary feed	12	7	12	350
2	Auxiliary feed	10	5⅝	12	250
1	Auxiliary feed	6	4	7	90

Feed Tanks.—There is in each forward engine room, a feed tank of a capacity of about 2,130 gallons. The tanks are fitted with an overflow valve and sight funnel, an automatic valve in the internal suction nozzle, and perforated partitions for the application of filtering material. There are also grease extractors in the main feed pipes of each group of boilers. These extractors are similar to large Macomb strainers, the basket or perforated internal chamber being filled with filtering material, by-pass valves and pipes and hinged cover bolts being provided to facilitate overhauling.

Steam Traps.—The separators, radiators, jackets, steam coils in the ship's coppers, the steam tables in the pantries, and the drains of the main and auxiliary steam pipes are fitted with Kieley

standard steam traps. These have been found to work very satisfactorily under exacting conditions.

Fire, Bilge and Water Service Pumps.—In each forward engine room, there is a vertical duplex double acting Blake pump, 12 x 7 x 12, which draws from the feed tanks, main condensers, sea, bilges, main and secondary drainage pipes and compartments, and delivers into the boilers, overboard and to the fire main. In each after engine room, there is a vertical duplex double acting Blake pump, 12 x 7 x 12, which draws from the crank pits, bilges, sea, main and secondary drainage pipes and indirectly from the compartments, and delivers into the fire main and overboard. In each engine room, there is a vertical duplex double acting Blake pump, 6 x 4 x 7, which draws from the sea and delivers into the water service pipes and the fire main.

Turning Engines.—In each engine room, there is a double cylinder vertical, simple engine, with cylinders 7 inches in diameter and a stroke of 7 inches, secured to the engine frame, for turning the main engines. It operates on a worm wheel on the line shaft through bevel gears and a worm. The worm is made to slide on a feather key, and is held in place by a collar below and a removable key above it. A double pawl ratchet is fitted to the shaft of this engine for turning by hand.

Other Auxiliary Machinery.—In each fire room hatch, there is a Williamson double cylinder ash hoisting engine, operated by a wheel outside of the bulkhead. Each engine is capable of hoisting 300 pounds from the fire room floor to the deck in five seconds, with a steam pressure of 60 pounds per square inch.

In each engine room hatch, there is a two-cylinder Sturtevant blower engine, with 4-inch cylinders and a stroke of 3 inches. The face is 48 inches in diameter. The blowers for ventilating the ship are situated, two in a compartment abaft the after engine rooms, and two forward above the protective deck, making four in all. Each is driven by a two-cylinder engine, with cylinders 5 inches in diameter and a stroke of 4 inches.

On the gun deck forward, there is a two-cylinder vertical windlass engine. On the spar deck, there are horizontal two-cylinder winches made of a special "low" pattern, so that the guns will clear them when the turret is turned.

Amidships, there is a boat crane on each side, each driven by a two-cylinder vertical engine. On the after bridge, there is a boat boom engine with two drums, one for the topping lift and the other for the hoisting purchase. This engine is also a two-cylinder vertical engine, driving the drums by worms.

Turret Turning Engines.—The forward and starboard turrets are turned by electricity, the port and after turrets, by steam. The steam turning engines are double vertical engines, with cylinders 8 inches in diameter and a stroke of 7 inches. Each engine has a 2½-inch piston valve, the travel being 1½ inches. The eccentrics are set at right angles to cranks. The crank shaft is 3 inches in diameter. The engines are secured to the bulkhead, and turn the turret tube by means of gears and worms. They are capable of turning the turret at the rate of one revolution per minute with the guns run out and the vessel heeled 10 degrees, with a steam pressure of 100 pounds per square inch. At a recent trial, the steam gear worked slightly better than the electric gear, but the result of the rivalry between the steam turned turrets and electrically turned turrets has no doubt been beneficial, as the present steam gear is believed to work much better than any steam turning gear that has been used in our Service.

The steam gear is worked by a lever in the sighting hood; this lever, by appropriate mechanism inside of the ammunition tube, moves a change valve on the engine. The "follow up" gear has been abandoned, and the automatic stop, which is necessary to prevent the turret going too far, is provided by cams which are fastened to the bottom of the revolving turret tube. As the turret nears its extreme position, the cams are brought to bear against a fixed arm connected to the valve gear. These cams close the valve gradually, and the turret will stop at the same point, regardless of the speed of rotation.

All ammunition hoists are electrical.

Telemotor.—To take the place of the wire rope or shafting which controls the steering engine, there is fitted an electrical steering telemotor. This electrical gear, in case of accident, can be thrown out.

Refrigerating Machinery.—An Allen dense air ice machine of a capacity of 1 ton per day is fitted. The scuttle butt water is kept cool and there are refrigerating rooms for officers and crew.

Workshop Engine.—The power tools in the engineer's workshop are driven by a vertical, single-cylinder engine; diameter of cylinder, 8 inches, stroke, 8 inches. The following machine tools are installed. One back geared screw cutting lathe with 24 inches swing over the ways and 16½ inches over the carriage, and taking 4 feet between centers. One 14-inch screw cutting lathe to take 40 inches between centers. One column shaping machine of 12 inches stroke and 30 inches traverse. One double geared drilling machine with screw feed, capable of drilling from ½ to 1½ inch holes and 12 inches from edge of work. One small drilling machine for hand and power, and a combined hand punch and shears.

Distilling Apparatus.—There are two distillers and two evaporators. The evaporators have straight tubes of brass which are expanded into the tube sheets; the back chamber is of cast iron and comes out with the tubes. The test of the evaporating plant showed its capacity to be 11,808 gallons per day. The temperature of the distilled water was 85, and of the sea water, 65 degrees Fahrenheit; the average steam pressure in tubes was 80, and in the shell, 34 pounds per square inch. The rated capacity of the plant is 10,000 gallons of potable water per day at a temperature of not more than 90 degrees Fahrenheit.

The following pumps are fitted in the evaporator room. (1) Single-acting evaporator feed pump, diameter of steam cylinder, 4½ inches, and of water cylinder, 2¾ inches, the stroke being 6 inches. (1) Duplex trap suction pump, diameter of the steam cylinder, 2 inches, and of water cylinder, 1¼ inches, the stroke being 2¾ inches. (1) Compound circulating and fire pump, diameters of steam cylinders, 6 and 12 inches, and of water cylinder, 9 inches, the stroke being 12 inches. This pump can be worked high pressure if so desired. (1) Tandem fresh water and brine pump, diameter of steam cylinder, 4 inches, and of each water cylinder, 2⅜ inches, the stroke being 5 inches.

Electric Plant.—For electric lighting, there are three generating sets made by the General Electric Company, Schenectady, N. Y.

In each set there is a 32-kilo-watt, compound wound dynamo of the multipolar type, having a capacity of 400 amperes at 80 volts. The engines driving the dynamos are two-cylinder vertical, inverted; diameter of cylinder, 12¼ inches, stroke, 5 inches. The crank shaft and armature shaft are connected by a rigid coupling. The engines are designed to run, with full load, at 400 revolutions per minute with 80 pounds steam pressure. The bed plate is common to both engine and dynamo.

For the motors turning the forward and starboard turrets, there is a special generating set, made by the same company, consisting of one double cylinder, vertical, inverted engine, with cylinders 12¼ inches in diameter and a stroke of 5 inches, and two 24-kilo-watt, multipolar dynamos on the same shaft. The switch boards are so arranged that the dynamos for the electric light service can be switched on to the turret motors in case of a break down of the turret dynamo.

There are 750 incandescent lights in the ship, nearly all of 16-candle power, a few of 32-candle power being used in signal and running lights. Of these 750 lights, 166 are in the engine and fire rooms.

There are four search lights having 30-inch projectors. They are mounted on stands, two forward and two aft. Each one of these lights requires a current of 90 amperes at 50 volts.

A small dynamo is fitted to take the place of the electric battery on bell circuits in case of failure of the latter. Telephones and voice tubes are fitted to connect the different parts of the ship with each other or to the central station. Electric telephones, engine telegraphs, range indicators for the guns, transmitters of orders and other electric devices are fitted.

WEIGHTS.

Weight of all machinery, including water, tons	1,543
reciprocating parts, H.P. piston, pounds	826
I.P. piston, pounds	1,476
L.P. piston, pounds	3,338
piston rods, pounds	2,849
connecting rods, pounds, one, 3,153.....three..	9,459
crossheads, pounds, one, 1,040.....three..	3,120

The bunkers are fitted with steam fire extinguishing pipes.

The following is a list of engines on board the *Brooklyn* with the number of steam cylinders.

4	Main engines,	12
4	Air pumps,	8
4	Circulating pumps,	4
2	Auxiliary condenser pumps,	2
4	Water service pumps,	8
4	Fire and bilge pumps,	8
4	Turning engines,	8
4	E. R. ventilating blowers,	8
7	Feed pumps,	14
12	Fire room blowers,	24
6	Ash hoists,	12
1	Ice machine,	1
1	Workshop engine,	1
4	Distiller and evaporating pumps,	6
1	Windlass engine,	2
2	Boat cranes,	4
3	Winches,	6
2	Turret turning engines,	4
1	Steering engine,	2
4	Dynamo engines,	8
4	Ship's blowers,	8
1	Oil pump,	2
2	Steam launch engines,	4
81	Total,	156

Much of the descriptive matter is taken from the report on the trial trip of the U. S. S. *New York*, by Passed Assistant Engineers E. R. Freeman and M. A. Anderson, U. S. N., which was printed in the JOURNAL on page 613 of Vol. V.

OFFICIAL SPEED TRIAL.

The official speed trial took place on Thursday, August 27, 1896, on the measured course off the New England coast, between Cape Ann and Cape Porpoise. The weather was fine and the sea smooth, making the conditions most favorable. The first run over

the course was made in 1 hour, 54 minutes and 42.52 seconds. The turn, made without change in the speed of the engines, occupied 20 minutes and 53.85 seconds. The return run was made in 1 hour, 52 minutes and 26.34 seconds. A tidal correction applied to the 83-mile course, reduced the latter to 82.953 nautical miles.

The machinery worked smoothly and without water on any journal, except that circulating through the bearings. Indicator diagrams were taken every half hour from each main cylinder, and once an hour from the main air and circulating pumps. No difficulty was found in keeping the steam pressure up to the desired point without running the blowers at too high a speed. The ease with which the steam pressure was maintained was no doubt due, in a great measure, to the high smoke pipes. All boilers were in use and under forced draft. The coal used was Pocahontas of good quality.

DATA OF TRIAL.

Draught, at beginning of trial, forward, feet and inches	21–6
aft, feet and inches	22–3
mean, feet and inches	21–10½
Displacement at above draught, tons	8,150
Area of immersed midship section, square feet	1,225
Wetted surface, square feet	29,500
I.H.P. (A), per 100 square feet of wetted surface	62.18
at 10 knots, reduced in ratio of 3.5 power	5.91
Average speed, knots	21.9117
Slip, mean of both propellers, per cent	18.67
Speed^3 × area immersed midship section ÷ I.H.P. (A)	702.55
Speed^3 × displacement^{⅔} ÷ I.H.P. (A)	232.26

SYNOPSIS OF STEAM LOG AND RATIOS.

	Starboard.		*Port.*
Revolutions per minute, main engines	136.2		136.9
mean of both engines		136.55	
Piston speed, feet, per minute	953.4		958.3
Steam pressure, boilers, per gauge		158.3	
engines, per gauge	158.5		157.4
1st receiver, absolute	79.9		78.2
2d receiver, absolute	30.7		31.2
Vacuum in condensers, inches of mercury	25.5		24.9
Opening of throttle		Wide.	
Cut-off in decimals of stroke from beginning, H.P.	.74		.74
I.P. and L.P.	.70		.70

	Starboard.		*Port.*
Double strokes of air pump, per minute	10.25		8.87
Revolutions of circulating pumps, per minute	169.7		175.4
Temperature in degrees Fahrenheit, engine room	100.3		100.2
fire rooms		118	
injection		62	
discharge	122.7		114.8
feed water		125.9	
Revolutions of blowers per minute, fire rooms (12)		590	
engine rooms (4)		401	
Air pressure in fire rooms, inches of water		2.26	
Mean effective pressures in cylinders, forward engine, H.P.	61.09		60.2
I.P.	33.59		29.9
L.P.	14.49		13.31
after engine, H.P.	56.49		62.47
I.P.	31.46		29.95
L.P.	14.09		14.61
equivalent reduced to L.P. piston, forward engine	40.53		37.4
after engine	38.32		39.32
both engines	39.42		38.36
I.H.P., forward engine, H.P.	1,382.89		1 372.43
I.P.	1,662.12		1,461.78
L.P.	1,694.38		1,565.14
after engine, H.P.	1,281.69		1,413 01
I.P.	1,557.24		1.489.66
L.P.	1,650.44		1.717.50
total, forward engine	4,739.39		4,399.35
after engine	4,489.37		4,620.17
both engines	9,228.76		9,019.52
collective, four main engines		18,248.28	
air pumps, two each side	8.21		8.34
circulating pumps, two each side	37.42		40.62
collective, all main engines, air and circulating pumps (A)		18,342.87	
feed pumps (6)		82.04	
blowers, forced draft (12)		269.06	
other auxiliaries		75.65	
total, all auxiliaries		426.75	
collective, main engines, air and circulating pumps, (B), all machinery in operation		18,342.87 18,769.62	
Ratio of volume swept by L.P. piston of one engine (mean) to that of the two buckets of one air pump, per minute	128.47		149.06
I.H.P. (B), per square foot of G.S.		18.47	
H.S.		.56	
cooling surface		.81	
per ton of machinery (total weight of 1,543 tons)		12.16	

MARINE BOILERS, PARTICULARLY IN REFERENCE TO EFFICIENCY OF COMBUSTION AND HIGHER STEAM PRESSURES.

By J. R. Fothergill, Vice-President.

[Read before the Northeast Coast Institution of Engineers and Shipbuilders in Cardiff, on June 25, 1896.]

As each year passes, the development in marine engineering so progresses that, at the present day, we have reached that high standard of efficiency which makes it difficult to realize how further economy can be obtained without a material increase in the boiler pressure.

Past experience shows that no essential development in the marine engine has taken place without considerable hesitation, time to perfect detail, and isolated trials proving beyond question such alterations were productive of economy. The introduction of the "compound," followed by the "triple," in each case took considerable time to develop before they became universal; the "quadruple" at the present day is quite the exception, and it is difficult to realize how it can become the accepted engine without a material increase in the working pressure. Not only engineers, but shipowners are much concerned as to the solution of this problem, and particularly, if we are to go to higher pressures, what is to be the accepted type of boiler; nor need we be surprised at this, when we consider the controversy taking place as to the various merits or demerits of the water tube type.

Is this increase in pressure to take place, or are we to accept 180 pounds to 200 pounds as the limit? Have we not at each marked increase in pressure over and over again heard the old argument, increased pressure means increased temperature, with all the attendant difficulties, repackings, oils, leaky joints, boiler troubles, &c., eloquently expounded, but of what avail? Is it not

an inherent characteristic of the engineer to go ahead? One might as well try to push back the tide as prevent the engineer in his march of progress. Material progress cannot be made without failures, and well that it is so, for experience so gained builds up, as it were, a solid foundation on which to establish ultimate success.

The greatest difficulty in the past has been the manufacture of suitable boilers to meet the requirements of increased pressure. Demand and necessity for high quality material has been met by the energy and vigorous activity of the steel maker. Capable machines, suitable and able to deal with any thickness of plate and diameter of rivet, are obtainable, and the furnace manufacturers have kept well abreast of the times. Although under these circumstances we can construct boilers with tightness and absolute safety to withstand any working pressure, yet we must not overlook the fact that a boiler has other functions than the mere certainty of being able to stand an enormous bottled-up pressure. It is of vital importance that a boiler should be considered an apparatus whose essential functions are:—Evaporation and combustion; the rate of evaporation depending upon the efficiency of combustion and heat generated.

We may reasonably assume increased boiler pressure looms in the near future, and although it is impossible to foresee to what pressures we may ultimately go, yet 250 pounds to 280 pounds may be considered within the bounds of probability; it thus becomes of the utmost importance to consider whether the present type of cylindrical return-tube boiler with circular furnaces is the most efficient as an apparatus for combustion, generation of heat, and transmission of that heat to the water. It is freely admitted this type of boiler has been, and is, up to certain pressures, the most simple, the most "get-at-able," and serviceable marine boiler of any type introduced; but does this hold good for higher pressures?

The important objections seem to be weight, thickness of plate, particularly such places as are heating surface, heavy close staying, and especially small diameter of furnaces. The writer is well aware small furnaces are occasionally used, but

this is usually due to some peculiar necessity, and not from choice. Neither the Board of Trade nor Lloyd's rules prevent the use of fairly large furnaces of the corrugated or ribbed types under high pressure, but the necessity for keeping the diameter of the boiler shell as small as possible, to avoid exceptional thick plates and additional weight, &c., necessitates the use of furnaces of small diameter.

All engineers of practical experience fully appreciate the efficiency of large furnaces, and this is readily understood if we consider what takes place when fresh coal is thrown on the fire of a furnace in active condition. The coal at once absorbs heat, which disengages the bituminous hydrocarbon constituents by volatilization, or, in other words, generates the gaseous portion of the coal by distillation. Until this process is completed the fixed carbon or coke is not consumed. Volatilization is the most cooling process in nature, due to the quantity of heat which is directly converted from the sensible to the latent state. It is, in particular, due to this cause that heavy firing is so disastrous, and should be carefully avoided. As the hydrocarbon gases are volatilized by the first application of heat and pass off, hydrogen having a greater affinity than carbon for oxygen, readily unites with oxygen to the production of water—steam. The carbon thus liberated must be supplied with its necessary proportion of oxygen to the production of carbonic acid, otherwise it will be precipitated in an extremely divided state and pass away as smoke, or, if from various causes the supply of air to the gases should be exceptionally small and badly distributed, carbonic oxide will be produced with serious loss in the amount of heat generated. For carbon consumed to carbonic acid evolves 14,500 units of heat, which, if consumed to carbonic oxide, would yield 4,400 only. Although carbonic oxide may be produced under exceptional conditions yet, fortunately, it does not appear from the analysis of funnel gases to be generally present, and when present it is usually in inappreciable quantities.

To ensure complete combustion to the generation of heat, it is essential that not only shall the air supply be sufficient, but that it shall be properly distributed in such manner as will

readily facilitate completion of the combustion of the hydrocarbon gases as they leave the surface of the fire, as well as the fuel on the bars. This question of air supply is one of considerable difficulty, and has exercised the ingenuity of many capable men. This difficulty is materially increased by reducing the diameter of the furnace, and thus the height above the fire.*

It is of material importance that the necessity for maintaining at high temperatures the hydrocarbon gases as they are evolved should be thoroughly understood. Hydrogen effectively unites with oxygen at not less than 600 degrees Fahr., and carbon with oxygen at not less than 800 degrees Fahr. The higher they exceed these temperatures within certain limits the more readily they unite and the less surplus oxygen required. Volatilization being the most cooling process in nature, it is imperative the greatest consideration be given to raising the temperature of the hydrocarbon gases as they are evolved, that they may immediately take up and unite with the necessary oxygen to complete combustion and generation of heat.

Perhaps in this respect the most serious difficulty to overcome is that great bugbear—heavy firing. Unfortunately, this appears inevitable where the manipulation of the fires is left to the capricious and prejudicial bias of the stokehold. Large quantities of coal thrown on the fire at once reduces the temperature, and frequently to such a degree that the gases evolved cannot take up the oxygen required, and thus they pass away, forming the huge volumes of smoke so commonly seen immediately after firing.

The smaller the furnace the greater the evil, in that the furnace crown is very much closer to the fire; in fact, without very careful and judicious firing—and where will you find it in the ordinary mercantile marine?—the surface of the fire is so close to the crown that immediately the gases are evolved they come in contact with the crown plate; and if we consider that at 250 pounds pressure, the temperature of the water, and hence the plate is, say, 400 degrees Fahr., we at once realize the deleterious effect this must have in keeping the temperature of the evolved gases

* See the writer's paper, "Combustion of Coal and Prevention of Smoke," read before this Institution, November, 1892.

far below that necessary to complete combustion. Again, the smaller the furnace the less the sectional area above the fire, and thus the velocity of the gases leaving the furnace require to be considerably increased; moreover, this velocity is greatly augmented by the necessity to largely increase the consumption per square foot of grate, due to the small grate area obtainable.

To obtain this necessary high velocity of the gases a lofty funnel with high temperature of the escaping gases must be used, or some system of induced draft. High velocity of the escaping gases is, in many respects, a most objectionable feature. Time is an essential element in the completion of combustion, in the union of carbon with oxygen, and in the transmission of heat to the water. At this high velocity, the temperature of the gases as they escape from the tubes into the uptake must be abnormally high. True, much of this heat may be utilized in various ways, but it is a costly process.

There is another important feature which does not appear to have attracted sufficient consideration, namely, the gases of complete combustion do not transmit heat by radiation, but by conduction. The sectional area through the tubes, being in all cases larger than the outlet from the furnace, facilitates the gases passing from the furnace at high velocity into the tubes in the most direct course without coming in close contact with the large heating surface of the combustion chamber; and probably the suction draft may be so great, when large quantities of coal are burnt per square foot of grate, as to draw the whole of the gases through the lower rows of tubes, leaving the top rows useless.

In Fig. 1, the writer has endeavored to depict the course gases of high velocity, due to excessive funnel or induced draft, will probably take when passing from the furnaces into the tubes, and from which it will be seen that practically a large percentage of the combustion chamber heating surface is rendered of little or no use, due to the inability of the gases to transmit their heat other than by conduction.

It would appear as though many of the objectionable features referred to are obviated in the water tube boiler. Of course

there are water tube boilers and water tube boilers, as there are in most other things, some to be entirely avoided, others specially designed and satisfactory for certain purposes only.

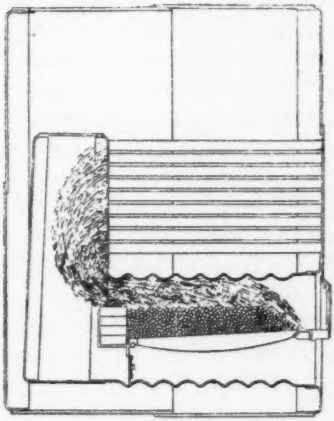


FIG. 1.

For the purpose of this paper, a type of water tube boiler known as the Babcock and Wilcox is given in Fig. 2. The writer has selected this particular boiler as one that he is somewhat acquainted with, and suitable to the requirements of the ordinary mercantile marine. He specially desires to say that he has not selected this boiler in detriment to other types of water tube boilers, but principally because he has evidence of its success, as fitted in several steamers, in confirmation of which he gives the following: Mr. Alexander Gray, consulting engineer, Newcastle-on-Tyne, who has under his supervision four steamers fitted with Babcock and Wilcox boilers, informs the writer they are giving the greatest satisfaction both as to consumption and endurance; in fact, the results are so satisfactory that they are specified to be fitted in two new steamers now building. The writer having received the following letter from Mr. J. Spear, superintending engineer to Messrs. Wilson, Sons and Co., Hull, ventures to give it *in extenso*, believing it will be considered most interesting:

"DEAR SIR: I believe we were the first in this country to try Messrs. Babcock and Wilcox's water tube boilers in a sea-going steamer; this was over three years ago in our S. S. *Nero*. This boiler is running quite satisfactorily, and the ship has not lost a day for boiler repairs up to the present. A few tubes in the feed water heater have been renewed, and probably the whole of them will require renewal shortly, as they appear to corrode very rapidly. In the boiler proper only one or two tubes exposed to the fire have been renewed at a trifling cost. About twelve months ago we fitted a new steamer, the *Hero*, with a pair of boilers by the same makers, but somewhat modified in details, no heater was fitted, and the tubes were larger in diameter. We have had considerable trouble with the bottom row of tubes in these boilers, and in consequence of this, in a third ship, the S. S. *Cameo*, which we have had fitted with Babcock and Wilcox boilers, we have reverted to the smaller tubes—slightly larger than the *Nero's*, and up to the present no tubes have given way, neither have any repairs been required to these boilers.

"As regards consumption there is not much either way; comparing the S. S. *Hero*, *Nero* and *Cameo* with our most economical forced draft and ordinary boilers, they fall slightly short, but still give as good results as ordinary averages with the ordinary boilers.

"I think the direction in which economy in consumption is to be sought for in the water tube boilers, is in the arrangement of baffles over or amongst the tubes, so as to properly utilize the heat that frequently too easily goes up the funnel.

"We have, as you are probably aware, one steamer fitted with four Belleville boilers. We have not been fortunate in our experience with these, but perhaps this has been from causes other than the design, although it is certain that the Belleville boiler is more difficult to clean and repair than is the Babcock and Wilcox, whilst with us the consumption has been greater with the Belleville than with the Babcock and Wilcox.

"Yours truly, &c."

In the Babcock and Wilcox type of boiler, we have a large grate area suitable to the best conditions for complete combustion.

The furnace is supplied with two doors sufficiently large to facilitate the distribution of the fuel, and at that convenient height which gives the fireman complete charge over his shovel, a matter of no small importance, particularly with untrained men. By firing at each door intermittently, you have always one side of the furnace incandescent when the other is being fired, and thus the hydrocarbon gases evolved are readily heated up to the necessary temperature to their union with oxygen to complete combustion and generation of heat. The height of the lower row of tubes above the fire offers a great advantage for the supply of air to the surface of the fire.

The writer is not aware whether the following arrangement has been tried, but it would appear to offer some advantages:— Say, for a distance equal to one-third the length of the grate, specially designed slabs of fire brick be suspended to the under side of the bottom row of tubes, or otherwise fixed. As the fire burns down and becomes a mass of incandescent coke, the fire-brick would store up a considerable amount of radiated heat, which in its turn would be given off in heating up the early gases as they are evolved to complete combustion and production of heat. Such an arrangement would necessitate placing the coal principally at the front of the fire and pushing it back on the grate after the gases were given off. It will be seen by the drawing, Fig. 2, that provision is made to deflect the gases in their passage through the boiler. This is one of the most important features of the water tube boiler, and due to this not being appreciated in the earlier types, was undoubtedly responsible for much of the loss of efficiency in those boilers, as the gases of combustion passed direct from the fire to the funnel without coming sufficiently in close contact with the heating surface. There is no doubt the most efficient position in which to place these deflectors is open to some argument. Whether they should be horizontal or vertical, viz., in a line with, or at right angles to, the tubes is debatable. Horizontal deflectors offer convenient shelves for the accumulation of soot and small ashes, &c.

The writer in his forced draft trials had strong practical evi-

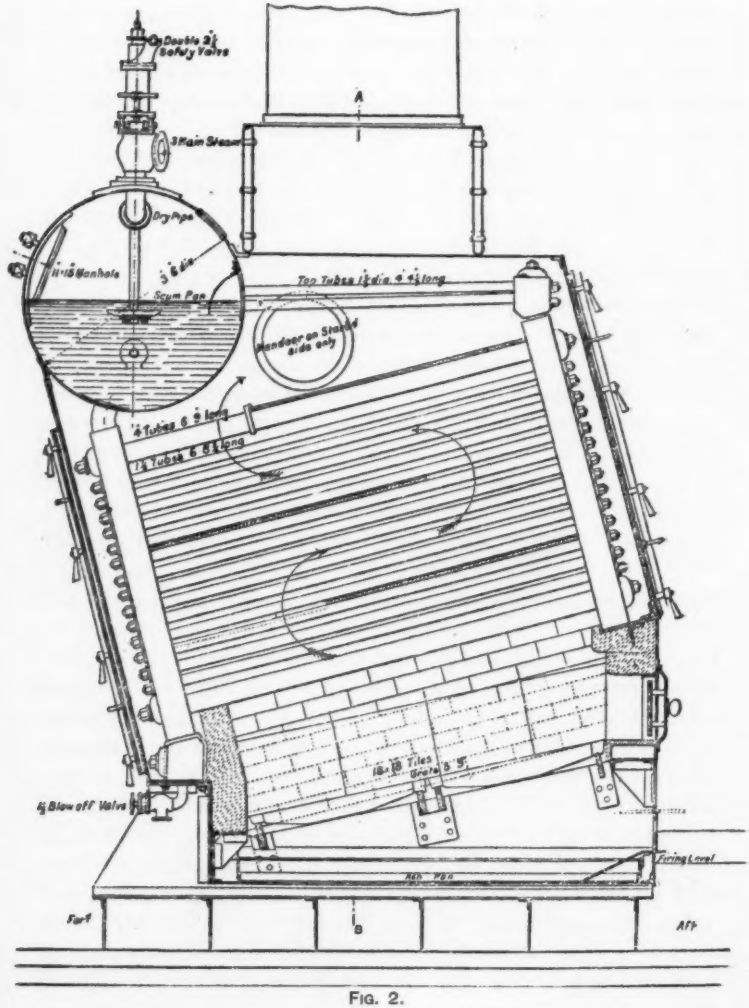

Double 2½ Safety Valve
3 Main Steam
Dry Pipe
Manhole
Scum Pan
Top Tubes 12 dia. 6' 4½ long
Handhole on Steam'd side only
4 Tubes 6' 9½ long
14 Tubes 6' 8½ long
4½ Blow off Valve
18 x 18 Tiles
Grate 5' 5"
Ash Pan
Firing Level
Fwd
A
B
Aft

FIG. 2.

dences of increased efficiency by keeping the gases passing through the boiler under slight compression. Take the case of the S. S. *Iona*, one of the steamers tested by the Committee of the Institution of Mechanical Engineers, and under the writer's supervision, and fitted with his system of forced draft. During the whole of the trial the damper in the funnel was only one-sixth open; this gave just sufficient compression to bring the gases into intimate contact with every square inch of heating surface, both in the tubes and in the combustion chambers, to the greatest advantage in the transmission of heat, as shown by the low temperature of the funnel gases. From frequent trials, all other things being the same, the difference in maintaining steam with the damper full open and only open one-sixth was most marked. It will be readily understood this closing of the damper cannot be utilized with natural draft. It is more than probable the application of some such system of slightly compressing the gases in water tube boilers would do much to improve their efficiency and obviate the necessity for deflectors and baffles, which in itself would be a considerable improvement.

In a paper of this limited description it is impossible to consider the mechanical structure and the many details necessary to an efficient boiler, and which must very largely determine the type of boiler of the future. There is no doubt the present type of boiler, if it does die, will die hard, due to its well-earned popularity, and in all probability it will only be the demand for higher pressures that will kill it. The writer feels he has left much unsaid, in fact, that he has only touched the fringe of this important subject, yet he hopes sufficient material will be found to promote useful discussion, for, after all, the greatest value of a technical paper is not so much the subject-matter it contains as the discussion it promotes and the ventilation of different opinions.

Discussion.

Mr. J. Gravell said it would be agreed that for higher pressures of steam than were used in the boilers of steamers at the present time, they would necessarily have to look forward to a

system of boilers which possessed strength, together with a moderate thickness of materials, and this led them unmistakably towards the adoption of some sort of water tube boilers. So far, the water tube boilers in use on board steamers had not been an unqualified success, especially as regarded economy in fuel, but this did not mean that the present system of water tube boilers could not be improved upon, and he had no doubt that means would soon be devised to remedy that defect and to make the water tube boilers as economical, or more so, than the ordinary boilers now in use.

Mr. W. Reavell said that the sea trials were expected soon of a third steamer, the largest vessel of the turret type yet built—the *Norman Isles*—which was also fitted with Babcock & Wilcox boilers. There were also two other turret ships in course of construction, which were also to be fitted with these boilers. With regard to that portion of the letter sent by Mr. Spear which referred to the heater in the boiler of the *Nero* it should, of course, be pointed out that the heater was not a necessary component part of the Babcock & Wilcox water tube boiler, but was fitted in this case for experiment. The type of heater used in the boiler of the *Nero* merely consisted of a series of tubes placed above the boiler itself, resting on the horizontal tubes, and so arranged that the water was pumped through the tubes of the heater on its way to the boiler. The trouble which was referred to in the letter occurred very soon after the boiler was first put to work, but this was remedied by the fitting of zinc rods inside the lower tubes of the heater. They considered that the fitting of zinc rods in feed water heaters was quite sufficient to make tubes in the heaters last as long as the tubes in the boilers. There was another point touched upon both in Mr. Spear's letter and Mr. Fothergill's paper to which he should like to refer. Engineers very frequently compared the water tube boiler with the Scotch type, *plus* forced or induced draft. That, he thought, was a little unfair, because any type of forced or induced draft added to a boiler of the Scotch type could be just as well added to the boiler of the water tube type. It should be judged on its

merits independently of the efficiency of the boiler itself. That was to say, if they wished to compare the types of boilers, it was only fair to take the boilers themselves without any forced or induced draft. Or if they wished to compare them with forced draft, let the same type be put on to each, and make the conditions exactly the same. In the latter part of his paper, Mr. Fothergill discussed the question of baffles, and expressed the opinion that all that was needed to make water tube boilers as satisfactory as the best type of Scotch boiler was to see that the baffles were so arranged as to get every part of the heating surface made effective, or else to have a slight compression in the furnace and around the tubes so that the gases touched every portion of the heating surface. The difficulty of ensuring that all the heating surface is used applies equally to boilers of the Scotch type; note for example the growing use of Serve tubes and retarders. The lines which Mr. Fothergill went on, with regard to the *Iona* were possibly the correct ones. Mr. Alexander Gray had also devoted some attention to this question. In some experiments on two turret ships with Babcock & Wilcox boilers all the baffles amongst the tubes were removed: in one case, restricting the draft area at the top of the boiler over the top horizontal tubes as much as possible, making it just sufficient to enable the boilers to give the steam freely that was required. In the other case, the baffles which were fitted on the tubes in different positions were increased, and then observations were taken to see which was the better method, with the result that they could not discover any difference in economical working. From the point of view of cleaning it seemed to be an advantage to remove the baffles, because they simply formed ledges for the accumulation of soot.

Mr. J. Nodder agreed with the author of the paper in that the present type of boiler intended to die very hard. The water tube boiler had not got everything its own way yet. In his paper Mr. Fothergill stated, "It will be seen by the drawing that provision is made to deflect the gases in their passage through the boiler." Most of them would be aware of some boiler tests

recently made at West Hartlepool. They were working at 255 pounds pressure, and were fitted with Serve tubes and retarders, working with the Ellis-Eaves induced draft. The results were very remarkable. Indeed, they represented the ideal condition of a pound of coal per horse power. That was something to be proud of, and it was arrived at by the means recommended by Mr. Fothergill of hindering the escaping gases from going into the funnel, and so taking the heat with them. The Ellis-Eaves system did that. When the water tubular boiler came up to this efficiency he would support it, and should be pleased to apply to it the Ellis-Eaves system of induced draft and Serve tubes.

Mr. Reavell, replying to a question, said the weight of a Babcock & Wilcox boiler was 50 per cent. less compared with an ordinary boiler doing the same work under similar circumstances.

The discussion was adjourned.

THE USE OF RETARDERS IN SCOTCH BOILERS.

By Jay M. Whitham, Esq., Member.

General: Spiral retarders, made of sheet iron and inserted in the fire tubes of marine boilers, have long been in use, and have proven of value in fuel saving. The writer, having completed a series of trials without and with retarders in a stationary boiler of the horizontal tubular type,* was successful in obtaining permission to conduct some further trials on the two Scotch boilers of the Steamship *Winyah* of the Clyde Steamship Company, plying between Norfolk and Richmond, Va.

EQUIPMENT OF STEAM PLANT.

Two Scotch boilers, 10 feet 3 inches in diameter by 11 feet long, each having two corrugated furnaces 42 inches in mean diameter. Each boiler had 148 tubes, 3 inches in diameter by 8 feet 1 inch long. The water heating surface per boiler was 1,051 square feet. The grates being 5 feet 10 inches by 3 feet 4 inches per furnace, the area was 38.9 square feet per boiler, or $\frac{1}{27}$th of the heating surface. The air openings in the grate were 46 per cent. of the entire surface.

Natural draft was used on the trials, the stack being 60 inches in diameter and 55 feet off the grates.

The triple expansion engine had cylinders 16, 24 and 41 inches in diameter by 30 inches stroke, and was connected to a surface condenser. The auxiliary pumps and engines usual for such steamers were operated on each test.

The retarders were made of No. 20 iron having a pitch of 6 feet, or $1\frac{1}{2}$ convolutions per tube. On the trial with the retarders one was placed in every other tube.

MANNER OF MAKING TRIALS.

The trials of the boilers, made without and with the retarders, covered periods of 8 and 9 hours respectively. The coal was

* An abstract of these trials follows this paper.

weighed, an inferior grade of Clearfield, Pennsylvania, bituminous coal being used. The engine was indicated every 15 minutes. The condensed steam discharged from the hot well to the filter box was measured, as was also the water drawn from the tanks to provide for leakages. All pressures and temperatures were noted at 15-minute intervals.

RESULTS OF TRIALS.

The results of the boiler trials without and with retarders may be summarized as follows:—

Items.	Without.	With.
Pounds of water evaporated from and at 212° F. per pound of dry coal	9.44	10.18
Boiler or evaporative H.P. developed	351.0	346.9
Engine H.P. developed	572.7	554.7
Square feet of heating surface to one boiler H.P.	6.00	6.06
Temperature of waste gases, degrees F.	655	585
Stack draft, inch	.43	.43
Water consumption in pounds from and at 212° F. per hour per engine H.P.	19.3	21.2
Pounds of dry coal consumed per hour per engine H.P.	2.24	2.12

From the above we note the following advantages in using retarders:

Reduction in stack gases, degrees Fahrenheit	70
Increased evaporation per pound of coal, pound	.74
Saving of coal per boiler H.P. developed, per cent	7.94
Saving of coal per engine H.P. developed, per cent	5.36

The difference between the last two items is due to the throttling of the steam at the engine, on the last test, thus reducing the effectiveness of the steam in the engine.

CONCLUSIONS.

1. Retarders, properly used, effect a saving in the use of coal.

2. Retarders check the draft, and their use is not advised except when it is strong.

3. Retarders can be best applied when the draft is strong and the gases are high in the stack.

"THE EFFECTS OF RETARDERS IN FIRE TUBES OF STEAM BOILERS."

[Abstract of a paper read by Mr. J. M. Whitham before the American Society of Mechanical Engineers, and forming part of Vol. XVII of the Transactions.]

The trials were conducted on a horizontal tubular boiler, rated at 100 horse power, at the Sutherland Avenue station of the Philadelphia Traction Company, Philadelphia. The purpose of the trials was to ascertain under what conditions, if any, retarders in the fire tubes would add to the efficiency of the boiler.

The boiler was 5 feet in diameter and 20 feet long, and contained 44 tubes, 4 inches in diameter. The grate surface of the boiler was 26.7, and the heating surface, 1,137 square feet.

On certain tests, retarders were used in the tubes. These were made of loosely fitting strips of No. 10 sheet iron, running the whole length of the tubes, and twisted to a pitch of ten feet, or making two entire convolutions in the length of the tube. The tests were conducted according to the methods advised by the American Society of Mechanical Engineers, and all instruments used were standardized. Measurements of the steam were made by a Barrus's Universal Calorimeter. The coal and water were weighed. Observations, at fifteen-minute intervals, were taken of the pressures and temperatures. The coal was practically uniform in quality on all the tests, the percentage of ash and refuse varying from 8.72 at the slowest, to 4.25 per cent. at the highest rates of combustion.

Variation of Economy with Capacity.—The tests showed that, by forcing, the 100-horse power boiler, rated on 11.37 square feet of water heating surface to the horse power, could easily develop over 200 horse power. At its rated power, 3.03 pounds of water, from and at 212 degrees Fahr., were evaporated per hour, yet on a forced test there was no difficulty in evaporating 7.26 pounds

to one square foot, or in getting a horse power on 4.76 square feet of heating surface.

The cost in fuel of a horse power was as follows for the tests made *without the use of retarders:*

Boiler horse power.	Pounds dry coal hour to one horse power.
52.4	3.30
74.6	3.21
99.7	3.25
125.3	3.30
150.0	3.24
169.6	3.22
199.7	3.36
217.4	3.51
239.0	3.82

These results show that there is practically no change in the economic workings of the boiler when run at from 50 per cent. below to 70 per cent. above its rating, *i. e.*, when making a horse power an hour on anywhere from 21.7 to 6.7 square feet of heating surface.

The cost of fuel per horse power was as follows for the tests made *with retarders in the tubes:*

Boiler horse power.	Pounds dry coal hour to one horse power.
52.4	3.30
77.3	3.21
104.2	3.15
127.5	3.18
148.6	3.14
169.1	3.11
197.3	3.23
226.1	3.23

It is therefore evident that *retarders enable a boiler to be run as economically on 5 square feet of heating surface to the horse power as on 21.7 square feet, or, practically, as on any number of square feet between these limits.*

CONCLUSIONS.

1. Retarders in fire tubes of a boiler interpose a resistance varying with the rate of combustion.

2. Retarders result in reducing the temperature of the waste gases, and in increasing the effectiveness of the heating surface of the tubes.

3. Retarders show an economic advantage when the boiler is pushed, varying in the tests from three to eighteen per cent.

4. Retarders should not be used when boilers are run very gently, and when the stack draft is small.

5. It is probable that retarders can be used with advantage in plants using a fan or steam blast under the fire, or a strong natural or induced chimney draft, when burning either anthracite or bituminous coals.

6. Retarders may often prove to be as economical as are economizers, and will not, in general, interpose as much resistance to the draft.

7. Retarders can be used only with fire tubular boilers.

8. The economic results obtained on the boiler tested are ideal, showing that it was clean, the coal good in quality, and the firing skillful.

With retarders the tubes are more effectively cleaned than without their use.

9. The tests prove that the marine practice of using retarders is good, and that the claim, often advanced, that they show from five to ten per cent. advantage, holds, whenever the boiler plant is pushed and the draft is strong.

TRIPLE SCREW VESSELS.

[Opinion of Professor Carl Busley, German Navy, in his paper on "Our Fleet," Zeitschrift des Vereines Deutscher Ingenieure, August 8, 1896.]

Much interest will be felt in the fact that the triple screw system is to be installed in all our new vessels now building, on the battleships as well as on the cruisers. The French have also adopted this system to a greater extent in their new vessels, the Italians and Americans have it already in use, and now the Russians are about to build triple screw vessels. Economic results are, unfortunately, not to be obtained by the triple screw system, as the Americans first found in their comparison of twin and triple screw vessels, and as we have now seen in our thorough trials of the *Kaiserin Augusta*. (The principal results of these trials were given on page 630 of the August number, but Mr. Busley's table, which contains some additional information and differs in some of the figures, is here given.)

As will be seen by the table, the maximum performance with three screws was 21.65 knots and 13,900 I.H.P., the slip of the middle screw being 10.49 and of the side screws, 13.8 per cent. With the center screw removed, the I.H.P. was 8,822 and the speed, 19.3 knots; the slip rose to 18.3 per cent. With the center screw replaced and then uncoupled, the speed, with almost exactly the same I.H.P., a difference of only 1 H.P., was only 18.8 knots, or ½ knot less. The slip was 19.4 per cent. After the two side screws had been removed and with only the center engine working, a speed of 15.2 knots was obtained with 3,948 I.H.P., the slip having now reached 28.4 per cent. The side screws having been replaced and then uncoupled, the speed was 14.6 knots with practically the same I.H.P., while the slip reached 32.5 per cent. This great increase in the slip is the cause of the rise in the coal consumption at service (?) speed.

When using the center screw alone, the coal per hour is 5,201 pounds, and with the two screws only, it is 5,075 pounds. The loss, due to the resistance of the two uncoupled wing screws, is thus shown to be greater than the advantage gained by the use of the center engine with full pressure over two equally large engines working at half pressure only.

For economy's sake it would, therefore, be better to steam such cruisers with two engines, instead of with only one, as is done for practical reasons.

At the lowest speed, the slip of the center screw rises to nearly 40 per cent., and its use alone becomes more inefficient.

If in spite of all this, triple screw vessels are building as has been shown above, there must be weighty reasons to justify such a course. This is, in fact, the case, and these reasons are, from the navigating, engineering and construction points of view, as follows:

The military or navigating reasons are that, as the blades of twin screws usually project far beyond the hull, these vessels cannot comfortably lie alongside of a dock or go through narrow dock gates, as the striking of the blades against the walls may easily happen. Besides, with these projecting screws, there is the danger that during an engagement, as for instance, a mishap in ramming or running past too closely, a propeller blade may be broken off. The center propeller of a triple screw vessel is, however, so well protected that it is not exposed to any danger of damage. Experience has furthermore taught that twin screw vessels, when one of the engines or propellers has been disabled by an accident, can only attain $\frac{3}{4}$ of their former greatest speed with the remaining engine, although the same full boiler power is available. But if a triple screw vessel loses one propeller, nearly $\frac{9}{10}$ of the greatest obtainable speed with three screws can still be maintained with the remaining two screws. Besides, the triple screw vessel will have greater handiness and manœuvring capabilities on account of the more favorable location of the rudder behind the center screw, an advantage of great importance in these days of fast steaming. In addition, a triple screw

vessel steaming with one engine, needs only one-ninth of its total engineer force to run it in three watches. Should such a cruiser suddenly be required to go into action, two-thirds of the engineer force, perfectly fresh and rested, would be available. This is a factor which cannot be too highly estimated, for the speed and manœuvring of the cruiser depend on it to a large degree.

The engineering reasons for the use of three screws may be summed up as follows: The extreme practicable limit for the size of one engine may be placed at about 15,000 I.H.P. This engine, to run easily at the necessary high piston speed, must have a very long stroke. But, a vertical engine of this description will be so high that it will project far above the protective deck and make protection impracticable. Three smaller engines, each of 5,000 I.H.P., which can easily be installed under the protective deck on account of their short but adequate stroke, are besides preferable, because they do not require such large castings and forgings as the single powerful engine. The requisite soundness of these parts, and, therefore also, their strength and durability, can certainly be more readily detected for the three small engines, and they can on this account be put in with greater assurance.

Finally, the advantages in construction follow from the fact that fast ships must have very fine lines aft, so that the water may follow to the propellers with the least resistance. The engines can be installed better in the sharp after body of three screw vessels than in similar twin screw vessels, for there is still ample room for the third engine in the narrow space abaft the other two engines. The whole machinery is moved further aft and the shafts may be made shorter. In bad weather, the shafts are, therefore, subject to less bending action than the long ones of twin screw engines. The shafts need not, therefore, be made so strong, and a saving of weight results. Finally, in the narrow vessels, the engine keelsons, to which the bed plates are secured, are closer together. The bed plates need not, therefore, be so strong, although offering the same resistance, and a further reduction in weight is effected.

Number of propellers.		Total I.H.P.	Coal burned per hour, pounds.	Speed, knots.	Slip of propellers, per cent.		Remarks.
					Center prop.	Wing props.	
(1) Maximum speed..	3	13,898	29,447	21.6	10.4	13.8	Center propeller removed.
	2	8,822	18,122	19.3	.....	18.3	Center propeller uncoupled.
	2	8,819	18,120	18.8	.....	19.4	Wing propellers removed.
	1	3,948	8,377	15.2	28.4	.....	Wing propellers uncoupled.
	1	3,897	9,524	14.6	32.5	.....	Starboard and center propellers uncoupled.
	1	4,491	8,267	14.1	32.5	.....	
(2) Service speed, ½ full power.	3	2,119	5,511	12.54	4.6	9.3	
	2	2,340	5,086	12.51	.....	17.0	Center propeller uncoupled.
	1	2,305	5,212	11.66	31.4	.....	Wing propellers uncoupled.
(3) Cruising speed.....	3	966	2,182	8.889	5.4	10.0	
	2	932	2,116	9	.....	17.0	Center propeller uncoupled.
	1	1,052	2,425	9.068	31.0	.....	Wing propellers uncoupled.
(4) Minimum speed...	3	356	816	5.362	4.9	9.5	
	2	296	683	5.5	.....	17.7	Center propeller uncoupled.
	1	450	1,014	5.732	39.0	.....	Wing propellers uncoupled.

NOTES.

NOTES ON DU TEMPLE BOILERS.

The following data of Du Temple water tube boilers have been taken from an article by Mr. V. Malfatti, engineer, Italian Navy, published in the "Rivista Marittima," Rome, September, 1896. The first three tests were made under full power; the results are the means of the observations recorded every fifteen minutes. No deduction for ashes has been made in the amount of coal consumed:

Shop number of boiler	210.		207.		220.	
Period	1st hour.	2d hour.	1st hour.	2d hour.	1st hour.	2d hour.
Steam pressure, pounds per square inch.	185	185	185	185	198	196
Air pressure in ash pit, inches	2.6	3.15	2.72	2.76	2.6	3.54
Air pressure in smoke pipe, inches	.94	1.18	.16	.16	.....	.....
Temperature in front of furnaces, degrees Fahr	72.5	75.2	82.4	95	76.1	71.6
Temperature at the back of boiler, degrees Fahr	102.2	102.2	91.4	154.4	107.6	103.1
Coal consumed per hour, pounds	1,323	1,323	1,323	1,411	2,205	2,557
Coal consumed per hour per square foot of grate, pounds	67.7	67.7	46.91	50.04	58.52	67.86
Water evaporated, pounds per hour	8,730	8,994	9,303	9,942	14,574	16,560
Water evaporated, pounds per hour per pound of coal	6.6	6.8	7.03	7.05	6.61	6.48
Water evaporated, pounds per hour per square foot H.S.	13.52	13.92	10.23	10.93	10.49	11.92

NOTE.—The temperature of feed water for the first boiler tested was 50° Fahr., for the other two 54° Fahr.

Boiler No. 210 was built in 1894 for the Norwegian torpedo boat No. 10, and was tested in November, 1895. The heating surface is 645.9, and the grate surface, 19.54 square feet. Ratio of H.S. to G.S., 33 to 1. The I.H.P. was about 665, and the coal per I.H.P. per hour, 2.0 pounds.

Boiler No. 207 was one of the boilers built for the French torpedo boats Nos. 77 to 86, and was tested on February 1, 1896.

This type is no longer used. The heating surface is 909.6, and the grate surface, 28.2 square feet. Ratio of H.S. to G.S., 32.26 to 1. The I.H.P. was about 718, and the coal per I.H.P., 1.9 pounds.

Boiler No. 220 was built in 1895 for a Russian torpedo boat, and was tested on February 17, 1896. The heating surface is 1,389, and the grate surface, 37.68 square feet. Ratio of H.S. to G.S., 36.86 to 1. The I.H.P. was about 1,232, and the coal per I.H.P. per hour, 1.93.

[NOTE.—As the I.H.P. itself was not given and had to be deduced from disagreeing ratios, all figures involved by the power are only approximately accurate, although the error will be small in any case.—EDITOR.]

The following table gives the weights and their relations to various factors.

Shop number of boiler	210	207	220
Weight, complete, without water, tons	4.77	5.69	7.87
of water, tons	1.06	.97	1.67
total, ready for steaming, with necessary accessories, tons	5.83	6.66	9.64
I.H.P. per ton of boiler, with water	114	107.8	127.8
square feet of grate surface	34	25.5	32.7
heating surface	1.03	.79	.89
cubic foot of space occupied by boilers	1.81	1.52	1.48

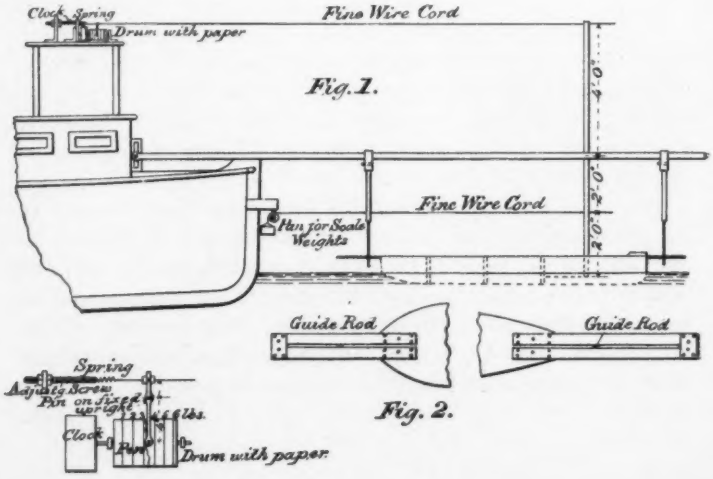
A further test was made with boiler No. 220 to determine its efficiency with the least combustion of coal. For this purpose, the grate surface was reduced to 13.56 square feet. The boiler was not covered with a non-conductor. The following are the average results of the two hours' test.

Steam pressure, 192 pounds per square inch; air pressure in ash pit, .04 inch; temperature in front of furnace, 78, and at back of boiler, 90° F.; coal burned per hour, 309 pounds; coal per square foot of grate surface per hour, 22.79 pounds; water evaporated, per hour, 3,141.5 pounds; water evaporated per hour per pound of coal, 10.17 pounds; water evaporated per hour per square foot of heating surface, 2.26 pounds.

APPARATUS FOR TESTING SHIPS' MODELS AT THE NAVAL CONSTRUCTION AND ARMAMENTS CO., BARROW-IN-FURNESS.

[From "Engineering," London.]

A new departure in the work of design was made at the establishment when the company were asked to construct torpedo boat destroyers of 27-knot speed, and the system organized by Mr. Adamson is specially interesting in view of its simplicity and the large measure of success that has been attained—the margin of accuracy comes well within the limits of practical success, and this, after all, is as much as can be said of most technical, as distinguished from laboratory, experiments. The system is a modification of that introduced by the late Mr. Froude; but the apparatus is inexpensive, and instead of an inclosed tank, the Devonshire Dock is utilized. The experiments were first made at Barrow in the winter of 1893, under the direction of Mr. George Brown, the assistant chief draughtsman, who has charge of the scientific work in the designing department. The data obtained were embodied in the design of the three 27-knot destroyers then built, and were amply confirmed by actual experience with the vessels when on trial. This success induced the builders to make similar experiments with the model of the later 30-knot destroyers.



The model is run with the assistance of a launch, the speed of the launch and of the model bearing a known relation to the speed of the full sized vessel, and the mechanism for ascertaining the resistance is placed on the launch. The method adopted and the recording apparatus are shown by figures 1 and 2.

It was, of course, inadvisable to tow the model behind the launch on account of the disturbance in its wake. A long spar or bowsprit was, therefore, rigged over the bow of the launch, the length of the projection being sufficient to keep the after end of the model about 4 feet in advance of the bow of the launch. The model is guided by rods at each end, these rods being free to swing fore and aft, while rigid in an athwartships direction. The rods hang from the spar into fore-and-aft guide slots of hard wood, projecting past the ends of the model. The necessity for these will be apparent to any one who has discovered the impossibility of towing a ship or boat in a straight fore-and-aft line without the help of the rudder or some other means of steering. A model free but for a towline fixed to its extreme fore end will, when towed, swerve to one side or the other indifferently, and will travel along in a half sidelong fashion.

The model is towed at the lower end of a long balanced lever, the connection being a wire link hooked to the lever, and to a bulkhead left in the model when originally hollowed out. The lever, left free to swing round its center in a fore-and-aft vertical plane, is pivoted at its center on the spar, and its upper end is connected to the recording gear by a fine wire cord, which previous to using is thoroughly stretched. Exactly half-way between the center of the lever and its lower end, another wire is connected to the lever and led aft over a pulley on the stern of the vessel. From the aft end of this wire is hung a scale pan to which is attached a permanent weight to keep the wires tight and put a slight initial strain on the spring. This helical spring is introduced into the upper wire, and its extension when the model is towed gives a measure of the resistance of the model. The extensions of the spring are recorded on an exaggerated scale on a paper stretched round a drum driven by clockwork. The markings are made by a small pen attached to one end of

a light wooden lever, the other end being fastened to the wire immediately forward of the spring. The clockwork is so arranged as to give about one revolution of the drum in the time taken to pass over the measured distance at full speed.

To obtain a scale by which to read the extensions of the spring in pounds resistance, weights of known specific gravity are placed in the scale pan at the stem of the model—two pounds in the pan being equal in effect on the spring and pen to one pound resistance of the model. By marking on the paper the position of the pen with different weights in the pan, a scale of pounds resistance was obtained.

When running, the clock is started, and as soon as the measured distance is reached, the pen is lowered on to the paper on the revolving drum, tracing an oscillating line. The oscillations are, of course, partly due to the natural oscillations of the spring, and partly to the vibration of the launch and the spar due to the beats of the propelling engines of the launch. The mean of the oscillations, obtained by integrating with planimeter, give the resistance of the model at that particular speed. The results are plotted in the form of a curve as resistance in pounds in terms of the speed of the model in feet per second. The spots, owing to the causes mentioned later, do not give a fair curve, but by taking plenty of spots, and running a mean curve through, the error is minimized.

The total resistance, of course, combined two elements—surface friction and wave making—the relations between these two for ship and model following a different law of comparison, and being in different proportions. Froude's experiments give a means of calculating the amount of surface friction resistance for both ship and model, the residue of the model's resistance being wave making, which, for corresponding speeds, varies directly with the displacement, or as the cube of the dimension. Dealing with the two parts separately, the total resistance of the ship is obtained, which, expressed in foot pounds per minute, gives the effective horse power, or horse power actually required to drive the vessel at the speed. To this has to be added an allowance for the power lost or otherwise used in driving pumps,

overcoming friction, etc., an amount which varies with the type and speed of engine and propellers, but in a case of this kind should not be much more than 40 per cent. of the total indicated horse power.

It is not contended that the results so arrived at are absolutely correct. With comparatively rough gear in an exposed dock, such consistent and steady results were not anticipated as are obtained with the elaborate and delicate machinery in use in covered-in experimental tanks. The aim was to furnish in the case of such unprecedentedly fast craft some independent and reliable check on the estimates for power, so as to avoid the risk of serious error. As a matter of fact, the results exceeded all expectations, particularly in view of the difficulties, and especially the fact that the model was run in a large dock in which the water is rarely absolutely smooth—a very slight ripple on the dock being, in proportion to the size of the model, a rough sea. It is, perhaps, unnecessary to remark that the experiments were only made in fine calm weather when the dock was as nearly as possible smooth. Again, the fact that no account is taken of the effect of the screw propellers on the vessel's resistance is in itself a serious drawback; but it gives a good independent check on estimates of power for exceptional speeds, and therein is great advantage.

The launch used has a maximum speed of just over 7 knots, which, with a model $\frac{1}{20}$th of the full size—the scale used in the first experiment—corresponds to a speed of $31\frac{1}{2}$ knots for the ship. As it was considered desirable in the case of the faster destroyers to have results up to a speed of about 35 knots, the scale of the new model was made $\frac{1}{24}$th of the full size running at 7 knots, this corresponds to a speed of $34\frac{1}{4}$ knots for the ship. Better results would probably have been obtained with a larger model, but the size was restricted by the speed of the launch used for towing the model. The speed of the launch was, for want of a better method, measured by the time taken to pass a known distance, marked on the side of the dock in the usual way by two pairs of posts set square off the course. The engines were set to the desired speed some time before reaching the

measured course, and not touched until the run was over, so as to have a constant speed on the run.

THE YARROW BOILERS.

Through the courtesy of Messrs. Yarrow & Co., Poplar, England, we are enabled to illustrate their automatic feed apparatus, fully described on pages 590 and 591 of the August number. A further description of this very simple apparatus, as fitted to the Yarrow boilers of the Argentine partially-armored torpedo boat *Santa Fè*, was given on page 609 of the same number. From all accounts, this apparatus has worked satisfactorily and is being fitted to all the boilers of the sister boats of the *Santa Fè*, the *Corrientes*, *Entre Rios* and *Missiones*. In the cut, the internal swivel steam pipe, and the arrangement for raising and lowering it, are clearly shown. The suggestion of regulating the feed supply by bringing the steam inlet to the donkey down to the water level, was made by Mr. Marriner, a member of the staff at the Yarrow works.

The JOURNAL is also indebted to Messrs. Yarrow & Co. for the cut of the three type boilers for the Dutch cruisers *Friesland*, *Holland* and *Zeeland*. As stated in the description of these boilers, pages 406–410 of the May number, Messrs. Yarrow & Co. built only three of the twenty-four boilers required for these cruisers, the others to be built in Holland from these three as patterns. Before leaving the works these type boilers were successfully tested to 350 pounds per square inch, 90 pounds above the Dutch government requirements.

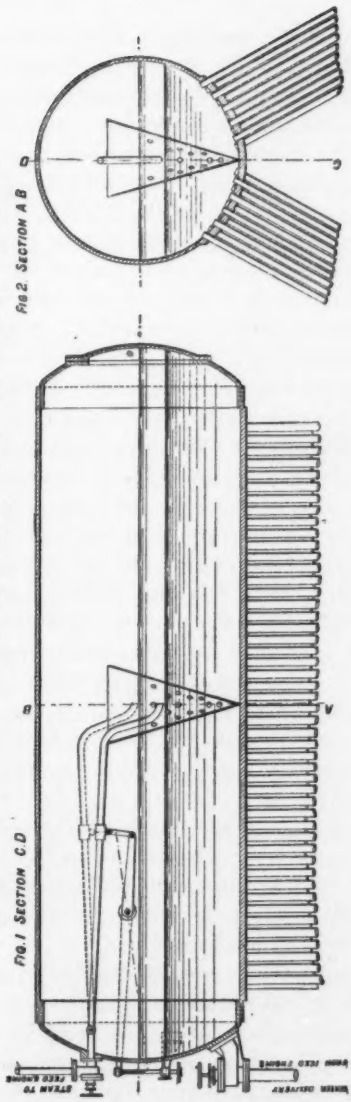
GALVANIZING PLANT AND ELECTROLYTIC PROCESS USED BY THE NAVAL CONSTRUCTION & ARMAMENTS CO., BARROW-IN-FURNESS.

[From "Engineering," London.]

The galvanizing bath is 17 feet 6 inches long, 5 feet broad, and 4 feet deep, containing 75 tons of molten zinc, maintained at a temperature of 600 degrees Fahr. Close upon 20 tons of material may be galvanized each week.

YARROW BOILERS FOR DUTCH CRUISERS "FRIESLAND," "HOLLAND" AND "ZEELAND."

YARROW'S AUTOMATIC FEED
AS APPLIED TO THE YARROW WATER-TUBE BOILER.



The electrolytic process, however, is preferred for a considerable part of the galvanizing work, as the temperature—70 degrees Fahr.—has no deleterious effect on the high tensile steel used, for instance, in torpedo boat destroyers, whereas in the hot bath, plates, especially when previously worked to the form of the ship, were inclined to buckle, and in the subsequent straightening the galvanizing is apt to scale off. Indeed, the Admiralty now specify that such high tensile plates must be galvanized by the cold or electrolytic process. The bath in this case has a strong solution of sulphate of zinc, and the steel is hung vertically between anodes of plates of zinc extending almost to bottom of the bath. The electric current is supplied from a 4,000-ampere dynamo, working at 5 volts, and driven at 700 revolutions by a 40-horse power gas engine. The bath is too far from the general boiler station in the works for convenient steam leads, and gas is more economical for this small power, the price being 2s. 9d. per 1,000 cubic feet. In both processes the plates are pickled in dilute hydrochloric acid—2 parts of acid to 1 of water—to remove scale and expose the bare steel. In the electrolytic process the plates have also to be scrubbed after pickling, and then dipped in a tank containing a strong solution of soda, so that any acid from the pickling tank clinging to the surface of the plates may be neutralized before the plate goes into the galvanizing bath.

The electrolytic process has no effect on the temper or set of plates, and thus they may be galvanized first and worked afterwards without the deposit scaling off.

The process is best adapted for plating, angles being only imperfectly coated in the bosom of the bar, and tubes only on the outside. The deposit cannot, in fact, take place on the surfaces which do not face the anode more or less directly. The more direct, the better the coating.

The temperature of the bath need never exceed 70 degrees Fahr., and immersion for about 30 minutes is, as a rule, sufficient to give the thickness of coating required by Admiralty specifications—.86 ounce per square foot of surface. Tubes for water tube boilers are galvanized by this process, of course only on the

outside. The influence on the preservation of the tube must be trifling, as the coating of zinc is probably soon melted off when exposed to the flame and hot gases in the boiler. But indirectly the process is useful. No deposit takes place in rents and cracks in the tubes, and such, even when too fine to be noticed, except microscopically, before galvanizing, are shown up clearly after immersion in the bath. In fact, tubes passed as sound and perfect before, have frequently been rejected after undergoing the process.

FULL POWER TRIALS OF THE BRITISH MEDITERRANEAN FLEET.

[From "The Naval and Military Record," London.]

Last Thursday, the 20th August, the fleet got under way shortly after 6 A. M. for the trip from Castellamare to Malta. Steam was raised for full speed to enable the quarterly four hours' full power natural draft trial and the 20 hours at three-fifths power to be run. As the cruisers required a distance of about 400 miles to accomplish these trials, they had to steam from Castellamare around the island of Pantellaria, and thence to Malta, the *Barfleur* and *Hebe* also going the long round, whilst the battleships went the course through the Straits of Messina. Shortly after leaving Castellamare, on the signal being made to go full speed, the majority of the ships commenced to work up for the four hours' trial, evidently with the idea of getting it over as soon as possible, and then setting down to the more comfortable three-fifths power. The ships soon commenced to "tail out," the *Sybille* forging ahead; but during the third hour of the trial this ship eased up, having to stop one engine, the stuffing box of the intermediate slide rod having cracked. Until that time she had been going 19½ knots, the remainder of the trip being done at 13. This is the second time the same mishap has happened to the *Sybille*, being due to excessive vibration, there being no guide brackets for the slide rods. Neither the *Cambrian* nor *Forte* did more than 18, though the *Forte* was a bit handicapped by having two defective fan engines; of the first class cruisers, the *Hawke* did far and away

the best. Taking several hours to work up to full power, by the time the other ships had finished, the *Hawke* then started and went away from everything, averaging 19.6 for the four hours, the *Theseus*, *Gibraltar* and *Vulcan* not doing nearly so well. Of the other ships which went the long route, the *Barfleur* did 16½ knots and *Hebe*, 16; of the battleships, the *Ramillies* did best, getting about 15½ knots. The *Scylla* and *Astræa*, which arrived at Malta a couple of days before the main body of the fleet, reported 19½ and 18½, respectively.

On the whole the trials were not a great success as regards speed, none of the ships, with the exception of the *Hawke*, *Scylla* and *Sybille*, obtaining their original trial speeds. This was no doubt due to want of docking, a good many of the ships, especially the Particular Service squadron, being badly in want of it. The whole of the ships arrived at Malta the following day.

The *Skipjack*, which had done no trial and left Castellamare the evening before the fleet, arrived first in the early morning, followed at half-hour intervals by all the remaining ships.

SHIPS.

UNITED STATES.

Alabama, Illinois and Wisconsin.—These battleships, named on October 28, were before known as *No. 8*, *No. 7* and *No. 9*, respectively. Bids for these vessels, described on page 596 of the August number, were opened at the Navy Department on Monday, September 14.

The following bids, all on the Department's plans and specifications, were received:

The Newport News Ship Building and Dry Dock Co., Newport News, Va., one ship for $2,595,000.

The Wm. Cramp & Sons' Ship and Engine Building Co., Philadelphia, Pa., one ship for $2,650,000, and two ships at the same rate.

Messrs. John H. Dialogue & Sons, Camden, N. J., one ship for $2,661,000.

The Union Iron Works, San Francisco, Cal., one ship for $2,674,950.

The Bath Iron Works, Bath, Me., one ship for $2,680,000.

The contract for building the *Illinois*, or *Battleship No. 7*, was awarded to the Newport News Co.; for the *Alabama*, or *Battleship No. 8*, to the Wm. Cramp & Sons' Co., and for the *Wisconsin*, or *Battleship No. 9*, to the Union Iron Works.

The designs of the Bureau of Steam Engineering having been adopted by all of the builders, the general data of the machinery are given here.

The engines, in two compartments, will be of the vertical, inverted, three-cylinder, triple expansion type, with cylinders 33.5, 51 and 78 inches in diameter, the stroke being 48 inches. The collective I.H.P. of propelling, and main air and circulating pump engines will be 10,000 when the engines are making about 120 revolutions per minute. The cylinders will be arranged in

regular order, the H.P. cylinder being forward. All main valves will be of the piston type, worked by the ordinary double bar link motion, there being one 14-inch H.P., two 20.75-inch I.P., and four 18-inch L.P. valves for each engine.

The engine framing will consist of wrought-steel columns trussed by wrought-steel stays. The bed plates will be steel castings of I-section, supported on the keelson plates. The crank shaft will be of forged steel in three interchangeable sections, and all shafting will be hollow. The shafts, piston and connecting rods, and all working parts generally will be forged of open-hearth steel, the thrust, line and propeller shafting having nickel added.

The condensers will be of composition and wrought iron or steel. Each main condenser will have a cooling surface of about 7,000 square feet, measured on the outside of the tubes, and the water will pass through the tubes.

Each main air pump will be double, vertical and single acting, worked by two steam cylinders. The two main circulating pumps will be independent and of the centrifugal type.

The propellers will be right and left, of manganese bronze or other equivalent metal.

There will be an auxiliary condenser, of composition, in each engine room, each with a cooling surface of 800 square feet, and a combined air and circulating pump.

There will be eight single-ended, four-furnace steel boilers, placed in four compartments. The total grate surface will be about 685, and the heating surface, about 21,200 square feet. The working pressure will be 180 pounds per square inch. Each boiler will be about 15 feet 6.5 inches in diameter and about 9 feet 11.25 inches long. The furnaces will be corrugated and have an internal diameter of 39 inches. There will be two smoke pipes. The closed fire room system of forced draft will be used, there being two blowers for each fire room.

One main feed pump will be placed in each engine room, and one auxiliary feed pump in each fire room.

Besides these, there will be the usual pumps for other purposes, steam reversing and turning engines, workshop en-

gine, a hydraulic pumping plant for various purposes, gun table or turret turning engines, distilling and evaporating plant, and other auxiliary machinery.

Oregon.—The following corrections should be made in the figures given on page 460 of the August number. Under "Totals for all boilers," the grate surface should be 616 square feet. Under "Ratios, all boilers," the first members of the four ratios should be 26.65, 30.44, .169 and 4.07, instead of 22.93, 26.19, .146 and 3.36.

The name of the writer of the paper on the trial of this vessel should be Leo Morgan.

Annapolis, Marietta, Newport, Princeton, Vicksburg, and **Wheeling.**—These composite gunboats, formerly known as *Gunboats Nos. 10, 15, 12, 13, 11* and *14*, respectively, were named on October 28. *Gunboats 10* to *13* are single screw vessels, and were described on pages 581 and 804 Volume VII. The *Annapolis* is building by the Crescent shipyard, Mr. Lewis Nixon, Elizabeth, N. J.; the *Newport* and *Vicksburg* by the Bath Iron Works, Bath, Maine; and the *Princeton* by Messrs. J. H. Dialogue & Son, Camden, N. J.

The *Marietta* and *Wheeling, Gunboats Nos. 15* and *14*, are twin screw vessels, and are building by the Union Iron Works, San Francisco, Cal. They were described on pages 583 and 804, Volume VII.

Torpedo Boat No. 3.—The first one of the three boats, *Nos. 3, 4* and *5*, the contract for which was awarded last year to the Columbian Iron & Dry Dock Co., Baltimore, Md., was successfully launched on October 1. The bids for these boats were given on page 152 of Volume VII.

These boats have a length, between perpendiculars, of 160 feet; beam, 16 feet 1¾ inches; depth, 10 feet 10 inches; draught, mean, normal, 5 feet; displacement at this draught, 142.5 tons. The normal bunker capacity is 9 tons, and the total, 44 tons. I.H.P., at 412 revolutions per minute, 2,000. Contract speed, 24.5 knots to be maintained for two consecutive hours. Weight to be carried on trial, 24.7 tons.

The twin screw engines are of the vertical, inverted, four-cyl-

inder, triple expansion type, with cylinders 12, 19.25 and 22 (2) inches in diameter, the stroke being 16 inches. Piston valves are used on all cylinders, there being one 6.25-inch valve for the H.P., two 7.25-inch for the I.P., and two 7.25-inch valves for each L.P. cylinder.

There are two water tube boilers, designed by the Bureau of Steam Engineering. They are in separate compartments, one forward of the forward engine compartment, and the other aft of the after engine compartment. The total grate surface is 95.6, and the heating surface, 5,260 square feet. The working pressure will be 250 pounds per square inch. The complement will be three officers and twenty men.

Torpedo Boat No. 6.—This torpedo boat, building by the Herreshoff Manufacturing Co., Bristol, R. I., and of which a few particulars were given on page 806 of Volume VII, was successfully launched on September 9.

30-knot Torpedo Boats Nos. 9, 10 and 11.—Bids for these boats, described on page 599 of the August number, were opened at the Navy Department on Friday, September 18. The various proposals were as follows:

The Union Iron Works, San Francisco, Cal., to build one boat of about 298 tons displacement, full load, for $227,500. The displacement on trial to be about 240 tons, including 35 tons of armament, coal, crew and equipment.

The Bath Iron Works, Bath, Maine, (*a*) two boats for $194,000 each, and three boats for $189,000 each. Displacement, full load, about 169.54 tons; on trial, 143.29 tons, including 20.29 tons of armament, &c. Also a bid (*b*) for two boats for $235,000 each. Displacement, full load, 288.9 tons; on trial, 226.65 tons, including 26.89 tons of armament, &c.

The Herreshoff Manufacturing Co., Bristol, R. I., one boat for $218,000, two for $209,000 each, and three for $206,000 each. Displacement, full load, about 286.4 tons; on trial, 232.14 tons, including 26.78 tons of armament, &c.

Messrs. J. H. Dialogue & Sons, Camden, N. J., (*a*) one boat for $260,000, and two boats for $255,000 each. Also a bid (*b*) for one boat for $275,000, and two boats for $270,000 each.

There was also an irregular bid of the Painton Electrical Steamship and Construction Co., Alexandria, Va., for one, two or three boats, to make 35 knots, for $265,000 each.

In October, contracts for these boats were awarded as follows: *Torpedo Boat No. 11* to the Union Iron Works, and *Torpedo Boats Nos. 9* and *10* to the Bath Iron Works on their bid (*a*) given above. *No. 9* will be very much like the British *Desperate*, built by Messrs. Thornycroft & Co., fully described on page 182 of the February number. *Nos. 10* and *11* will be practically copies of the French *Cyclone*, building by Messrs. Normand & Co. The *Cyclone* will be a modified and improved *Forban*, and is designed for 32 knots and 3,500 I.H.P.

Torpedo Boats Nos. 12 to 18.—Bids for these boats, described on page 601 of the August number, were opened at the same time and place as given above for the 30-knot boats. The bids were as follows:

The Columbian Iron Works, Baltimore, Md., (*a*) one, two or three 20-knot boats for $49,000 each on the Department's plans. (*b*) One, two or three 20-knot boats on their own designs for $45,000 each. (*c*) One, two or three 22.5-knot boats on their own designs for $79,100 each. The following combination bids on their own designs were also made: (*d*) One 20-knot and one 22.5-knot boat, $124,100 for both. (*e*) One 20-knot and two 22.5-knot boats, $203,200 in all. (*f*) Two 20-knot and one 22.5-knot boats, $169,100 in all.

Messrs. Moran Bros. & Co., Seattle, Wash., (*a*) one 20-knot boat, on the Department's plans, for $59,840; two boats for $58,675 each; and three boats for $56,928 each. (*b*) One 20-knot boat on their own designs, for $58,240; two boats for $57,075 each; and three boats, $55,328 each. (*c*) One 22.5-knot boat on their own designs, for $99,216; two boats for $96,239 each; and three boats for $94,255 each. They also proposed to build, on their own designs, (*d*) one 20-knot and one 22.5-knot boat for $155,953 for both. (*e*) One 20-knot and two 22.5-knot boats for $248,212 in all. (*f*) Two 20-knot and one 22.5-knot boats for $211,234 in all.

The Bath Iron Works, Bath, Maine, on their own designs, (*a*)

one boat of about the same size as the Department's design of 20-knot boats, but to have a speed of 22.5 knots, for $85,000; two boats for $81,000 each; and three boats for $80,000 each. (*b*) One boat of about the same size as the Department's design of 22.5-knot boats, but to have a speed of 25 knots, for $131,000; two boats for $125,000 each; and three boats for $123,560 each.

The Herreshoff Manufacturing Co., Bristol, R. I., on their own designs, (*a*) one 20-knot boat for $40,000; and two or three boats for $37,500 each, with $1,500 extra for each boat for galvanizing. (*b*) One 22.5-knot boat for $84,000; and two or three boats for $80,000 each, with $3,000 extra for each boat for galvanizing. (*c*) One, two or three boats, exact duplicates of the *Cushing* built by this firm, for $92,500 each. (*d*) One boat, same design as the *Cushing*, but built on the specifications submitted with bid *b*, for $82,500; and two or three boats for $78,500 each. They also proposed to build, on their own designs, (*e*) one 20-knot and one 22.5-knot boat, $120,750 for both. (*f*) One 20-knot and two 22.5-knot boats, $198,750 in all. (*g*) Two 20-knot and one 22.5-knot boats, $157,000 in all.

The Wolff & Zwicker Iron Works, Portland, Oregon, (*a*) one 20-knot boat, on the Department's plans, for $49,000; two boats for $47,500 each; and three boats for $45,000 each. (*b*) A duplicate of the *Cushing*, one 22.5-knot boat, for $83,800; two boats for $79,500 each; and three boats for $75,100 each. On their own designs, (*c*) one boat for $85,839; two boats for $81,546 each; and three boats for $77,150 each.

The Providence Steam Engine Co., Providence, R. I., (*a*) two 20-knot boats on the Department's plans, for $51,600 each; and three boats for $48,650 each.

Mr. Lewis Nixon, the Crescent Shipbuilding Co., Elizabeth, N. J., on his own designs, (*a*) one 20-knot boat for $69,000; and two boats for $68,000 each.

The Charles Hillman Ship and Engine Building Co., Philadelphia, Pa., on the Department's plans, (*a*) one 20-knot boat for $48,500 and two boats for $48,000 each.

Messrs. J. H. Dialogue & Son, Camden, N. J., on the Department's plans, (*a*) one 20-knot boat for $69,250 and two boats for

$68,500 each. On their own designs, (*b*) one 22.5-knot boat for $120,000 and two boats for $115,000 each.

Messrs. George Lawley & Son, South Boston, Mass., on their own designs, (*a*) one 20-knot boat for $61,500, two boats for $57,625 each, and three boats for $56,835 each. (*b*) One 22.5-knot boat for $83,500, two boats for $80,000 each, and three boats for $78,800 each. For a larger size, (*c*) one 22.5-knot boat for $102,000, two boats for $98,000 each, and three boats for $97,000 each. They also proposed to build on their own designs, (*d*) one 20-knot and one 22.5-knot boat of the larger size, $162,250 for both. (*e*) One 20-knot and two 22.5-knot boats, $256,000 in all; two 20-knot and one 22.5-knot boats, $215,500 in all.

Owing to the many and complicated bids, the contracts for these torpedo boats were not awarded until the early part of October, the result being as follows:

Torpedo Boats Nos. 12 and *13* to Messrs. Wolff & Zwicker on their bid (*c*), two 22.5-knot boats for $81,546 each.

Torpedo Boats Nos. 14, 15 and *16* to the Herreshoff Manufacturing Company on their bid (*g*), one 22.5-knot boat and two 20-knot boats, total cost, including galvanizing, $163,000.

Torpedo Boat No. 17 to the Columbian Iron Works on their bid (*b*), one 20-knot boat for $45,000.

Torpedo Boat No. 18 to the Charles Hillman S. & E. B. Co. on their bid of one 20-knot boat for $48,500.

Total contract price of these seven boats, $419,592; cost of ordnance outfit, $59,220; total cost, $478,812.

Walter Q. Gresham.—This steel revenue cutter, fully described on page 561 of Volume VII, was successfully launched at the Globe Iron Works, Cleveland, Ohio, on Saturday, September 12. The general data are here repeated. Length over all, 205 feet; length between perpendiculars, 188 feet; beam, 32 feet; depth, amidships, 17 feet; displacement on a mean draught of 10 feet 9¾ inches, 906 tons; I.H.P., about 2,000, when the engine is making about 160 revolutions per minute; cruising speed, 16 knots.

ARGENTINE REPUBLIC.

Garibaldi.—This armored cruiser, purchased in Italy, and described on page 605 of the August number, has finished her

trials. According to the "Rivista Marittima," Rome, the name of the vessel is *Garibaldi* and not *José Garibaldi.* The following data are from the same source.

The official trials were made over the new measured mile established by the Italian government off Genoa, preliminary trials over the old base off Spezia having shown that the water, which was about 65 feet deep, was too shallow. The depth of water on the new base is from 160 to 230 feet. Owing to bad weather and the numerous gun trials, the speed trial did not take place until July 6. Six runs over the mile gave a mean speed of 19.948 knots. The mean I.H.P. was 13,200, and the revolutions per minute, starboard, 104.5, port, 104.2.

Entre Rios.—The second of the four partially-armored destroyers, building by Messrs. Yarrow & Co., Poplar, completed her official trials on September 1. A complete description of the sister vessel, the *Santa Fé*, was given on page 609 of the August number. The trial of the *Missiones* is given below; the last of the remaining four destroyers, the *Corrientes*, has not been tried. The bunker capacity of these vessels is about 80 tons of coal. There are three smoke pipes. The crew is berthed forward, the officers' quarters are between them and the engine room, and the petty officers have quarters aft. The following account and data of the official trial are from "Engineering," London.

Draught of vessel, taken at Gravesend, forward, 4 feet 2 inches, aft, 4 feet 10 inches, mean, 4.5 feet. 20 tons of coal were on board, the total trial load being 35 tons. The vessel reached the Maplin measured mile about 3½ hours after leaving the Poplar yard, and the three hours' trial was at once commenced, the weather being fine.

Means for the six mile runs. Steam pressure in boilers, 147, in 1st receiver, 64, and in 2d receiver, 10 pounds per square inch; vacuum, 25 inches; air pressure in forward fireroom, 1.5, in after fire room, 1.75 inches; revolutions per minute, 365.2; speed, 26.803.

The average number of revolutions for the three hours was 364.8, giving a mean speed of 26.75 knots, or ¾ knot in excess of contract. The slip of propellers was 7.06 per cent.

It will be noticed that the steam pressure was only 147 pounds, while the designed working pressure of the boilers is 220 pounds per square inch. "Messrs. Yarrow and Co. apparently consider that if they carry out their contract with a good margin on the right side it is not desirable to aim at the utmost the vessel can do, simply for the sake of an empty boast. In this they are wise, for the vessel will be none the less capable of her highest speed in war because she has not been pushed to it in the official trials. If it were a case of speed premiums, we should see quite a different course followed by this firm."

The **Santa Fé** reached St. Vincent safely on her way to Buenos Ayres. She encountered the full force of the gales prevalent during the latter part of September, but proved herself a good sea boat.

Missiones.—The third of the partially-armored destroyers built by Messrs. Yarrow & Co., successfully completed her official trials on October 9. With the contract load of 35 tons, a mean speed of 27.1 knots was attained, exceeding the trial speed of her two sisters, the *Santa Fé* and *Entre Rios*.

Corrientes.—The last of the Yarrow destroyers was successfully launched on October 10.

AUSTRIA.

Magnet.—This torpedo cruiser, built by Mr. F. Schichau, Elbing, Germany, completed satisfactory trials and was accepted in July. The trial lasted three hours, and the speed, with forced draft, was 26 knots, 2 knots over the contract requirement. The vessel was fully loaded.

The following particulars are from "Mittheilungen aus dem Gebiete des Seewesens," Pola, Austria:

Length between perpendiculars, 221.4 feet; beam, extreme, 26.9 feet; load draught, mean, 10.82 feet; corresponding displacement, 502 tons; coal capacity, 102 tons.

The armament consists of six 1.85-inch rapid fire guns and three 17.72-inch deck torpedo tubes.

The twin screw engines were designed to develop 5,918 I.H.P.

There are four boilers arranged in two compartments, each pair of boilers having one smoke pipe.

Viper.—The JOURNAL is indebted to Messrs. Yarrow & Co. for the annexed half tone of this sea going torpedo boat built by them, and which has two of their water tube boilers. A description of the boat and machinery, and satisfactory results of the trials, were given on page 384 of the May number.

Natter.—This torpedo boat, built by Mr. F. Schichau, Elbing, and mentioned on page 385 of the May number, has completed her trials.

The length is 150.5 feet; beam, 17.38 feet; draught, load 8.79, feet; displacement, 150 tons. There is one engine, designed for 2,268 I.H.P., and two Thornycroft boilers. The designed speed was 23.5 knots. Coal capacity, 29.5 tons. The armament consists of two 1.85-inch rapid fire guns forward and three torpedo tubes, two forward and one aft. With a load of 25.8 tons, an average speed of 26.5 knots was obtained on the three hours' trial.

BRAZIL.

Barrozo.—This steel, sheathed cruiser, built by Messrs. Armstrong, Mitchell & Co., at their Elswick yard, Newcastle, was successfully launched on Tuesday, August 25.

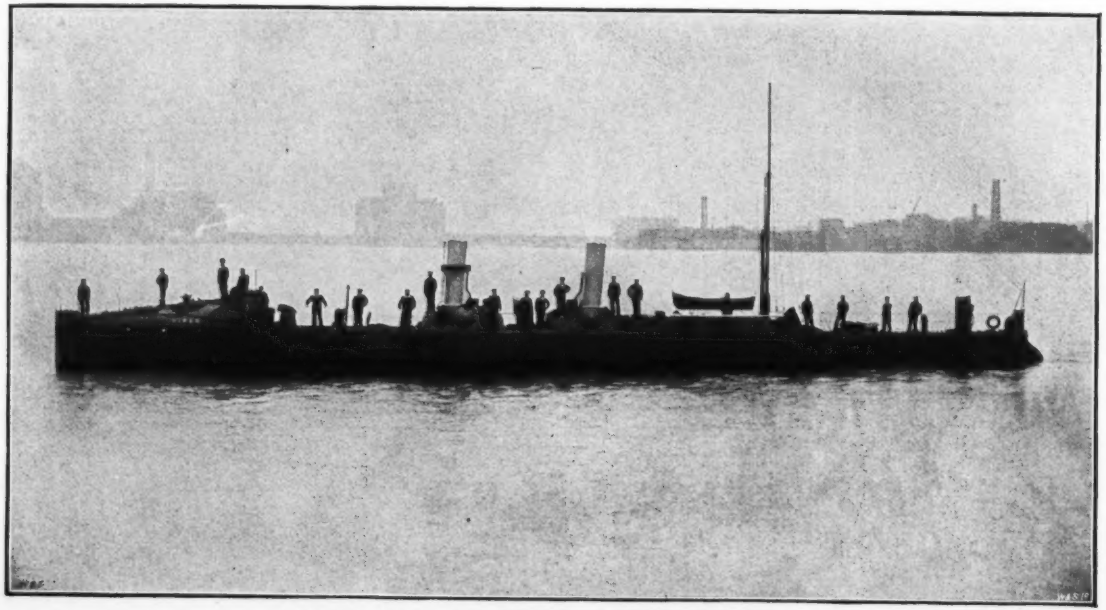
The length of the vessel is 330 feet; beam, 43.75 feet; draught, 16 feet 10 inches; displacement, 3,450 tons.

There is a curved steel protective deck extending the whole length of the vessel. It is 1¼ inches thick on the horizontal part and 3½ inches thick on the sloping parts.

The armament will consist of six 6-inch and four 4.7-inch rapid fire guns, all 50 calibers long and of the Elswick pattern; ten 6-pounder and four 1-pounder Nordenfeldts, four Maxims, two field guns and three torpedo tubes. Of the 6-inch guns, three are arranged to fire ahead and three astern.

The twin screw engines are designed for 7,500 I.H.P. and to give the vessel a speed of 20 knots.

The bunker capacity is 800 tons, the side bunkers extending to 6 feet above the water line.



AUSTRIAN TORPEDO BOAT "VIPER."

BUILT BY MESSRS. YARROW & CO., POPLAR, ENGLAND.

CHILI.

Capitan Orella.—This destroyer, built by Messrs. Laird Bros., Birkenhead, and mentioned on page 386 of the May number, completed her official full power trial satisfactorily on September 30.

The vessel's length is 213 feet; beam, 21.5 feet; depth, 12.75 feet; displacement, about 300 tons. The water tube boilers are of the Normand type, modified. There are four smoke pipes.

Six runs over the measured mile at Skelmorlie gave a speed of 30.17 knots, the revolutions per minute being 361. The average number of revolutions per minute for the three hours was 362.5, giving a speed of 30.23 knots. The I.H.P. was 6,200. Although there was a choppy sea, the vessel was very steady.

Capitan Muniz Gamero.—A sister vessel of the *Capitan Orella* described above, completed her official trials on October 15. The mean revolutions for the three hours were 364 per minute, and the speed slightly over 30 knots; for the six mile runs the revolutions were 369, and the speed, 30.42 knots.

ENGLAND.

Albion, Canopus, Goliath, Glory and **Ocean.**—These are the first class battleships of this year's programme, mentioned on page 378 and 612 of the current volume. The *Albion* will be built by the Thames Iron Works, and the machinery by Messrs. Maudslay, Sons & Field; the *Canopus*, at the Portsmouth dockyard, and the machinery by Messrs. Scott & Co., Greenock; the *Goliath*, at the Chatham dockyard, and the machinery by the Messrs. J. Penn & Sons, Greenwich; the *Glory*, hull and machinery, by Messsrs. Laird Bros. & Co., Birkenhead; and the *Ocean*, at the Devonport dockyard, and the machinery by Messrs. Hawthorn, Leslie & Co., Newcastle.

"Engineering," London, says that "Structurally they will be similar to the *Majestic* class, but to secure the reduction in displacement certain modifications have had to be made, the most significant being a reduction in the thickness of the harveyized armor which protects the broadside for the greater part of the length and for some 18 feet of its depth. The new ships will

have 6-inch plating instead of the 9-inch in the *Majestic*, but the armament will be the same, four 12-inch 45-ton guns being mounted *en barbette*, and these guns by reason of their great length—35.43 calibers—give, with an 850-pound shot, a muzzle energy of 33,940 foot-tons. There will also be twelve 6-inch quick firing and 30 smaller guns." There will also be five torpedo tubes, four of which will be under water. The cost of the guns alone is placed at $340,000.

The twin screw, triple expansion engines of these vessels are to have cylinders 30, 49 and 80 inches in diameter, the stroke being 51 inches. They are to develop 13,500 I.H.P. with 100 revolutions per minute, the speed being 18 knots, maintained for eight hours. The H.P. and I.P. valves will be of the piston type. There will be twenty Belleville boilers, with a grate surface of 1,100, and a heating surface of 32,000 square feet. The working pressure in boilers will be 300 pounds per square inch, reduced to 250 pounds per square inch at the engines. Feed water heaters will be fitted in the uptakes. The cost of the engines and twenty Belleville boilers and their fittings for each dockyard vessel is said to be about $500,000. The contracts require that the vessels shall be subject to a 30 hours' trial with 2,500 I.H.P., a 30 hours' trial with 9,500 I.H.P. and an 8 hours' trial with 13,500 I.H.P.

The total cost of the dockyard vessels is given as about $4,250,000. It is expected that all of these battleships will be completed near the end of 1898.

Victorious.—This first class battleship, a sister vessel of the *Majestic*, and described on page 813 of Volume VII, has successfully completed her official trials. This vessel was commenced at the Chatham dockyard on May 28, 1894, and launched on October 19, 1895. The machinery was built by Messrs. Hawthorn, Leslie & Co., Newcastle-on-Tyne. The total cost of this vessel will be between $4,250,000 and $4,500,000. The contract required the usual three trials, the 8 hours' natural draft trial, at 10,000 I.H.P., to be made with not over ½ inch air pressure, a 4 hours' forced draft trial, and a 30 hours' coal consumption trial at 6,000 I.H.P.

On August 29, a satisfactory preliminary trial of 3 hours' duration was made.

The first official trial, natural draft, 8 hours, was made on August 31; the forced draft trial on September 2, and the coal consumption trial on September 4 and 5. The results are as follows.

Eight hours' natural draft trial.—Steam pressure, 146 pounds per square inch; vacuum, 26.7 inches; revolutions per minute, starboard, 99.9, port, 99.3; average air pressure, .26 inch; I.H.P., starboard, 5,098, port, 5,218, total, 10,316; speed by log, 16.92 knots; total coal burned, 191,000, or, 2.31 pounds per I.H.P. per hour. The draught of vessel was 24.5 feet forward and 26 feet 10 inches aft, or 25 feet 8 inches, mean.

Four hours' forced draft trial.—Steam pressure, 147 pounds per square inch; vacuum, 26.5 inches; revolutions per minute, starboard, 107.6, port, 103.1, average, 105.35; air pressure, 1 to 1.25 inches; I.H.P., starboard, 6,139, port, 6,062, total, 12,201; speed by log, 18.7 knots. The draught was 24 feet 3 inches forward and 26 feet 1 inch aft, or 25 feet 2 inches, mean.

Thirty hours' coal consumption trial.—Steam pressure, 145 pounds per square inch; vacuum, 27.4 inches; revolutions per minute, average, 86.9; air pressure, *nil;* I.H.P., total, 6,205; speed by log, 14.9 knots; coal per I.H.P., 1.6 pounds; draught, mean, 25 feet 1 inch.

The vessel was finished at the Chatham dockyard October 17, and will soon be ready for commission.

Prince George.—This first class battleship, a sister vessel of the *Majestic*, and which was launched at the Portsmouth dockyard on August 22, 1895, has completed her official trials. The machinery was built by Messrs. Humphrys, Tennant & Co., Deptford. The description of the *Magnificent*, given on page 161 of Vol. VII, will apply to this vessel. For the electric lighting, there are three Brotherhood's compound double-acting engines, each set coupled direct to a Siemens' ironclad dynamo and supplying 600 amperes at 80 volts at a speed of 300 revolutions per minute, with a steam pressure of 100 pounds to the square inch.

The 8 hours' natural draft trial was made on September 8 with

the following results. Steam pressure, 150.7 pounds per square inch; vacuum, starboard, 25.7, port 25.5 inches; revolutions per minute, starboard, 96.9, port, 97.3; I.H.P., 10,464; coal per I.H.P. per hour, 2.3 pounds; speed by log, 16.52 knots; air pressure, 0.44 inch. The draught of vessel was 24 feet 11 inches forward and 25 feet 2 inches aft, or 25 feet ½ inch, mean.

The 4 hours' forced draft trial was made on September 14. Steam pressure, 152.3 pounds per square inch; vacuum, starboard, 26.4, port, 26.3 inches; revolutions per minute, starboard, 101.56, port, 101.98; I.H.P., starboard, 6,104, port, 6,149, total, 12,253; speed by log, 18.3 knots; air pressure, 1.2 inches. The draught was 24 feet 9 inches forward and 25 feet 3 inches aft, or 25 feet, mean.

The 30 hours' coal consumption trial was made about September 18. Steam pressure, 137 pounds per square inch; vacuum, starboard, 27.8, port, 27.5; revolutions per minute, starboard, 83, port, 82.8; I.H.P., 6,211; speed by log, 14.76; coal per I.H.P. per hour, 1.7 pounds. The draught was 24.5 feet forward and 25 feet aft, or 24.75 feet, mean.

Cæsar.—This first class battleship, a sister ship of the *Majestic*, was floated out of dock at the Portsmouth dockyard on September 2. The description of the *Magnificent*, given on page 161 of Volume VII, will apply to this vessel. The machinery was built by Messrs. Maudslay, Sons & Field. Torpedo nets will be fitted. The total coal capacity is given as 2,250 tons, or 400 tons more than any of the other ships of this class. Forced draft will be used.

Illustrious.—This first class battleship, a sister ship of the *Majestic* and the last of the nine battleships built under the Spencer programme of 1893–'94, was floated out of dock at the Chatham dockyard on September 17. The weight at the time of "floating out" was about 5,000 tons. The first plate was laid on March 11, 1895. The machinery was built by Messrs. John Penn & Sons, Greenwich. Induced draft will be used, as in the *Magnificent*, the machinery of which was built by the same firm. A description of the *Magnificent* was given on page 161 of Volume VII.

Majestic.—This battleship, fully described on page 167, and the trial results given on page 812 of Volume VII, has been fitted with new propeller bosses, the old ones being cracked through. Since this refit, says "Engineering," London, she has carried out the series of progressive trials that were ordered last March, but which have been deferred from time to time for a suitable opportunity. The first series was carried out in Stokes' Bay on October 14, with the following results: Revolutions, 24.46 starboard, and 24.57 port; total I.H.P., 223.04; speed, 4.103 knots. Revolutions, 61.07 starboard, 59.72 port; I.H.P., 2,224.27; speed, 10.357 knots. Revolutions, 73.66 starboard, 75.59 port; I.H.P., 4,116.20; speed, 12.547 knots. Revolutions, 87.03 starboard, 84.58 port; I.H.P., 7,010.48; speed, 14.305 knots. Revolutions (full power, natural draft), 98.52 starboard, 97.83 port; I.H.P., 10,082.9; speed, 16.096 knots. On Saturday, October 17, in fine weather, but with a strong wind on the beam in both directions, the ship carried out three runs with forced draft on the measured distance of 23 miles between Rame Head and Dodman Point. At each run the speed was taken by observation, with the following results: First run.—Air pressure, 1 inch; vacuum, 26.3 inches starboard and port; revolutions, 103.3 starboard, 102.9 port; I.H.P., 11,814; speed, 16.86 knots. Second run.—Air pressure, 1.08 inch; vacuum, 26.5 inches starboard, 26.3 inches port; revolutions, 102.5 starboard, 102.4 port; I.H.P., 11,914; speed, 16.9 knots. Third run.—Air pressure, 1.17 inches; vacuum, 26.5 inches starboard, 26.6 inches port; revolutions, 103.6 starboard, 103.2 port; I.H.P., 11,673; speed, 16.8 knots. Mean of the three runs.—Air pressure, 1.08 inches; vacuum, 26.4 inches starboard and port; revolutions, 103.1 starboard, 102.8 port; I.H.P., 11,800; speed, 16.85 knots. The speed of the ship was somewhat retarded by the heavy sea and wind, but it will be seen that under the imposed conditions, forced draft gave an additional speed of little more than three-quarters of a knot.

Terrible.—The official trial programme of this first class cruiser, as mentioned on page 613 of the August number, was commenced on August 5. Sixteen out of the forty-eight boilers were used. The coal burned was Cardiff hand-picked. There

were about 600 men on board, the firemen being nearly all trained men from the training squadron. The published data are few, perhaps owing to the restriction mentioned in the London "Times" that "the Admiralty have directed that none but the officers professionally engaged are to witness the steam trials of the *Terrible*."

The steam pressure was from 220 to 250 pounds per square inch; revolutions per minute, 64.5; I.H.P., 5,110. These are the averages for the thirty hours. Four runs over the measured mile in Stokes' Bay gave a mean speed of 13.43 knots. The total quantity of coal burned for 24 hours, omitting the three hours at the beginning and end of the trial, was 141 tons, or, at the rate of 5.875 tons per hour, or, 2.57 pounds per I.H.P. per hour.

The second trial, at 18,000 I.H.P. for thirty hours, was commenced August 7. After running for some time, the engines having attained 98 revolutions per minute, and developing about 19,000 I.H.P., the corresponding speed being about 19 knots, the trial was interrupted by "an accumulation of pressure inside of the vacuum ring of the S.I.P. slide valve," according to the London "Times." The vessel came to anchor in Plymouth Sound. "An inspection showed that the I.P. valve and cylinder were scored, besides other defects, but the cause of the trouble was not discovered. Later the vessel steamed to the Portsmouth dockyard at 50 revolutions." ("Times" and "Naval and Military Record," London.) The "Engineer's Gazette," Newcastle, states that "so far as can be ascertained, the intermediate valve seat, the intermediate slide face, and the cylinder liners were worn, it is said, by water finding its way into the cylinders."

Erratum.—In the August number, the previous description of the *Terrible* was incorrectly referred to page 584 of Vol. VI. It should have been Vol. VII.

Powerful.—This first class cruiser has satisfactorily completed part of her official trials, which were the same as mentioned on page 613 of the August number for the sister vessel, the *Terrible*. The preliminary contractors' trial of this vessel were given on the same page.

To the detailed description of the *Terrible* (page 584, Vol. VII) and to the points in which the *Powerful* differed (page 599, Vol. VII), may be added the following information from "Engineering," London, from which the trial data are also taken.

The keel of the *Powerful* was laid on March 10, 1894. She was in an advanced state, with all her wood sheathing and many of the internal fittings, when launched on July 24, 1895, and she was delivered to the Admiralty on July 14, 1896, arriving at Portsmouth on July 17. Since then she has been docked and copper-bottomed, her engines have passed through their turning trials in dock, and she is now practically ready for commission. The total cost of the vessel, ready for war, is given in the Navy Estimates as $3,576,895.

The total heating surface of the 48 Belleville boilers is 69,453 square feet, equal to 2.77 square feet per designed horse power (25,000 I.H.P.), and the total grate surface, 2,192 square feet.

The boiler installation occupies 186 feet of the length of the vessel, this length being divided into eight compartments by a longitudinal and five thwartship bulkheads. In each of the four after compartments there are eight boilers. These are fired fore and aft. In each of the four forward compartments there are four boilers. Owing to the "fineness" of the ship's lines, and the desirability of coal protection, it was decided to arrange those boilers to be fired athwartship. This secured the maximum boiler power for the area available, but it has been suggested that in the event of the rolling of the ship altering the inclination of the tubes, the efficiency of the boilers might be impaired, and without entering into the question at issue, it is interesting to note that in subsequent ships all the boilers will be fired fore and aft.

Twenty-four of the boilers have eight elements, and 24 seven elements, each containing 20 tubes, or, properly speaking, 10 pairs of tubes. These vary in length, being approximately 6 feet 7 inches. All are of lap-welded mild steel manufactured by Messrs. A. J. Stewart and Clydesdale. Owing to accidents having happened abroad to tubes by the opening or splitting of the weld, the Admiralty have decided that, hereafter, all tubes

for Belleville boilers are to be solid drawn. As this necessitates a special plant, only three or four firms have undertaken the manufacture of drawn tubes, and the cost of the tubes will be much increased.

It may be interesting to give here the conditions exacted from the tube manufacturers. The steel for the lap-welded tubes is made by the basic open-hearth process, and a strip of the metal annealed was subjected to a tensile strain of from 21 to 24 tons, the elongation being 25 per cent. in the length of 8 inches. Pieces cut from the tubes were subjected to a tensile stress of 26 tons, the extension being at least 20 per cent. in 8 inches. The other tests were: Strips cut from the tubes, flattened, heated to a blood heat, and plunged into water 82 degrees Fahrenheit, should be capable of being doubled over a radius of $\frac{1}{2}$ inch without fracture; pieces 2 inches long cut from the ends of tubes under $\frac{3}{16}$ inch thick are to be capable, when cold, of being hammered down endwise until the length is reduced to 1 inch, and of being flattened until the sides are close together, without fracture. The ends of the tubes under $\frac{3}{16}$ inch thick are to admit of being expanded by a three-roller expander to increase the diameter to 12½ per cent. in the case of a cold tube, and 20 per cent. with a hot tube. Tubes over $\frac{3}{16}$ inch thick are to be expanded to one-half this increase. These tests were applied to about 2 per cent. of the tubes, selected at Messrs. Stewart and Clydesdale's works by the Admiralty officer. The tubes were also tested by water pressure to 1,000 pounds. This in the case of the solid-drawn tubes has been increased to 1,500 pounds. This latter, however, is the only change in the tests for the solid-drawn tubes. The tube junction boxes are malleable cast iron, about $\frac{5}{16}$ inch thick, and were subjected to a water test of 750 pounds per square inch, and a drop test from a height of 25 feet on to a cast iron slab.

The main steam pipes, principally of lap-welded mild steel with riveted flanges, dip towards the forward end of the boiler compartments, with the object of draining back into the boilers any water lodging in the pipes. Each group of eight boilers is connected by a separate range of piping to a stop valve on the engine

room bulkhead. A connection from these valves leads to a steam separator in each engine room. As care requires to be exercised in starting the engines supplied with steam from Belleville boilers—there is a tendency to priming if they are suddenly rushed—this steam separator has been designed to avoid any difficulties due to water in the steam. It is formed of a vertical cylinder, about 8 feet long and 4 feet in diameter, made of riveted wrought iron plates, fitted with baffle plates and an automatic and hand drain apparatus. For the same purpose, there are fitted on the main steam pipes automatic drain traps of the well-known Admiralty pattern. Between the separator described and the main stop valve is the Belleville steam reducing valve, to which is fitted a safety valve loaded to 220 pounds, or 10 pounds above the working pressure in the engine.

The feed pipe to each boiler is $\frac{3}{4}$ inch in diameter.

Experiments have shown that the proportions of the feed pipes adopted in the Belleville system increased the pressure on the pipes by from 400 pounds to 600 pounds, and this has necessitated the ratio of steam cylinder to water cylinder in the feed pumps being made 2.4 to 1, as against 1.6 to 1 in ordinary practice. The main pumps—of the Weir type and numbering six—have a $16\frac{1}{4}$-inch steam cylinder and a $10\frac{1}{2}$-inch water cylinder, the stroke being 18 inches. Three of these are in each engine room, while distributed in the boiler compartments are four auxiliary pumps of the same size, and four with $13\frac{1}{2}$-inch steam cylinders and $8\frac{3}{4}$-inch water cylinders, with a stroke of 15 inches.

In order to prevent an undue pressure on the discharge side, an ingenious arrangement has been designed by Messrs. Weir, called a controlling gear, which automatically stops the working of the feed pumps, should the pressure in the discharge pipes exceed a predetermined maximum. The gear consists of a steam cylinder, on which there is always steam at boiler pressure, and a water cylinder in communication with the discharge feed pipes. When the pressure in the latter exceeds the normal amount determined, it acts on the area of the plunger and overcomes the pressure on the steam cylinder, which gives, or shuts off, steam to the main feed pumps.

Two of the boilers were worked at Barrow for some time to test the evaporation, feed, and safety valves. The table gives the results of two evaporative trials.

	No. 1.	*No. 2.*
Calorific value of coal, pounds of water per pound of fuel...	13.75	13.75
ash, pounds of water per pound of fuel....	.5	5
Percentage of ash and clinker.....	11.52	5.45
Water evaporated, in pounds.....	14,458	18,580
Coal consumed, in pounds.....	1,645	2,195
Coal consumed, in pounds per square foot of grate.....	18	24
Water evaporated per pound of coal, pounds.....	8.78	8.472
Water evaporated from and at 212 degrees, pounds.....	10.81	10.420
Steam pressure, 8 element boiler.....	260.6	260
7 element boiler.....	253.15	258
Air blowing pressure, inches.....	6.3	8
Temperature of funnel, degrees Fahr.....	631	639.7
Vacuum in funnel, inches.....	.46	.5
Temperature of feed water, degrees.....	42	42
Maximum pressure in feed pipes, pounds per square inch....	435	420
Heat units transmitted per square foot of heating surface per minute.....	99.34	127.6
Heating surface per pound of coal per hour, square feet.....	1.76	1.32
Efficiency of boiler.....	.786	.755
deducting ash.....	.792	.757
pump, vulcanite rings.....	.924	.92
Water evaporated per square foot of heating surface per hour, pounds.....	4.986	6.411

The evaporative trials clearly proved the necessity of careful stoking. The grate area is so large that there is the possibility of part of it being without fuel, and thus cold air rushes through, considerably reducing the temperature. In the early trials, before the workers at Barrow became accustomed to the boiler, the evaporation was fully 25 per cent. less than in the trial results recorded, and Mr. Durston is to be congratulated on his foresight in training stokers in the *Sharpshooter* for the larger vessels. On the *Sharpshooter*, the coal consumption decreased from 2.5 to 1.65 pounds per horse power hour.

Experiments were also directed to ascertain the efficiency of the safety valve; these resulted in modifications being made in the original design which prevented the accumulation being more than 10 pounds, while the valve returned to its seat with a loss of only 2½ pounds below the working pressure.

The air pumping engines, for supplying the air through nozzles to the furnaces, are of the Belleville type, with water jacketed air cylinders. The diameters are $7\frac{1}{4}$ inches and $13\frac{1}{2}$ inches, the air blowers being 14 inches in diameter. The stroke is 9 inches. They work at a pressure varying from 5 to 15 pounds to the square inch, according to the rate of combustion desired.

The smoke pipes are 80 feet above the fire. There is a large space between the top of the boilers and the underside of the protective deck. This is somewhat unusual in the design of warships, and is due to the less headroom required by the Belleville boiler. The result is a better atmosphere and more comfort to the stokers. There are 12 fans for supplying air to the stokeholds, each 90 inches in diameter, the engines driving them having 7-inch cylinders and 6-inch stroke.

The drains from the main and auxiliary steam pipes, stop and safety valves, and the auxiliary machines of the ship are led by a special service of pipes to the collecting tank in the engine room, from which the water is pumped to the feed tanks. This special drainage system, which was introduced by the Admiralty a few years ago, is a most important factor in reducing the loss of fresh water to the boiler.

The engine rooms are 60 feet long—the width of each varies from 18 to 21 feet—and the height from bottom of crank pit to protective deck is 23 feet. The feed tanks and hot wells are built into the vessel. On the after bulkhead, under the protective deck, in each engine room, there is a two-crank steering engine.

The cylinder diameters are 45, 70 and 76 (2) inches, the stroke being 48 inches. The sequence of cranks is H.P., forward L.P., I.P., after L.P., the first and third, and the two L.P. being at right angles. The cylinders are connected to each other by cast steel struts, passing through brackets cast on the sides of each cylinder. When the engines were first under steam, it was found that these struts expanded nearly one-half inch, and an allowance was made for this when setting them permanently. They are further stayed by two steel wire ropes, which are carried from the athwartship bulkhead, and secured by shackles to each side of the H.P. cylinder.

The cylinders are of cast iron, 2 inches thick, except the L.P., which are 1.75 inches thick. The tensile strength of this cast iron is about 9.5 tons per square inch, with a transverse strength such that a bar 1 inch square, placed on supports 12 inches apart, will carry 21 hundred weight at the center. The liners are of hard, close-grained cast iron with a tensile strength of 12.3 tons per square inch, and a transverse strength equal to carrying a load of 30 hundred weight as above applied. The pistons are conical, of cast steel, with two packing rings of Perkins' metal for each H.P., and broad cast iron rings for the other pistons. All packing is set out by spiral springs.

All the working parts are of forged mild steel. The piston rods are 11 inches in diameter. Each crosshead has two journals, each 13 inches in diameter and 13 inches long. The connecting rods are 13 inches in diameter and 7.75 feet long, or only 1.9 times the stroke. This short connecting rod was necessary to get the engines under the protective deck—there is only clearance for the trolley for the cylinder heads. Armor bars are placed in the three engine room hatchways. The forward and after hatches act as downcomers for the fresh air, while the middle one is fitted with two suction blowers.

The crank shaft is in four interchangeable parts, 20 inches in diameter with a 10-inch hole, and 46 feet 8 inches long over all. The crank pins are $24\frac{1}{8}$ inches long, and 22 inches in diameter with a 14-inch hole. The eight crank shaft bearings are each 23 inches long. The thrust shafts are 18.5 inches in diameter, each with nine horseshoe rings, the total surface for ahead thrust of each engine being 2,619 square inches. The line shafting is 18.5 inches in diameter with a 10-inch hole, and is supported by two bearings, each 23 inches long, and having white metal in the bottom half. The inner bearing in the stern tube is 4.5 feet long, and the outer 6 feet, both being $22\frac{3}{8}$ inches in diameter. The length of the outer bush on the A-bracket is 8 feet and the diameter 22.25 inches. The propeller shaft is 20 inches in diameter. The length of each shaft, from the after crank shaft coupling to the center of the propeller, is 117 feet 6.5 inches, making the total length of shafting for each engine 164 feet 2.5 inches.

The propellers are three-bladed, of gun metal, 19.5 feet in diameter and 24 feet pitch, and a developed surface of 75 square feet; they work inwards. The center of each propeller is 14 feet from the center line of the vessel and 15 feet forward of the after perpendicular. The center lines of the shafting extended would meet the forward engine room bulkhead 8.75 feet from the center line of the vessel. The bulkhead is 6 feet 9.5 inches forward of the ends of the forward crank shafts.

The main valves are on the forward side of each cylinder. The original arrangement of the starting valves on the cylinders was to admit steam at the top or bottom of the pistons, but it was found by tests that the consequent movement of the engine was rather slow, and modifications were made admitting steam to the I.P. and L.P. receivers.

After this modification, the engine was reversed from full ahead to full astern, without vacuum in the condenser, in from 14 to 18 seconds, while formerly it seldom required less than 45 seconds.

The filter tanks are fitted with a series of vertical perforated plates, placed in grooves, and having between them a thick Turkish toweling, on which the grease is deposited. As the feed and hot well tanks are close to each other, an overflow was easily arranged from the latter into the former. The feed tanks have a combined capacity of 35 tons.

The controlling gear of the main feed pumps was arranged on trial to stop the steam supply when the pressure in the feed discharge pipe reached 600 pounds per square inch.

The electric lighting plant consists of two vertical engines, with cylinders 10 inches and 17 inches in diameter by 10 inches stroke, coupled to dynamos which, running at 300 revolutions, develop 600 amperes at 80 volts. There is a third dynamo on the upper deck, the total number of lights being 1,000, with six search lights.

There are two distilling condensers and two steam pumps, divided into two independent sets of equal power, working compound with great economy of boiler steam consumption. Each of the four Normandy evaporators may be worked separately.

Automatic feed and steam regulators are fitted so as to prevent the possibility of salting up, and the evaporating surfaces can easily be removed for cleaning. The evaporators are capable of producing from sea water per 24 hours 120 tons (26,880 gallons) of fresh water, 25 tons thereof being for drinking purposes and the remainder for boiler feed.

In the workshop on the main deck, there are two lathes arranged in line, so that a head stock being removed the two beds may be used in conjunction for an extra long job. There is a third lathe, a combined punching and shearing machine, a shaping and planing machine, a drilling machine and other tools, driven by shafting worked by a two-cylinder engine.

The engine complement numbers 281, including the chief engineer and seven engineer officers as assistants, 18 chief and other engine room artificers, 9 chief stokers, 27 first and second-class leading stokers and 219 stokers. The ship's complement numbers 894, of all ranks and ratings.

On September 23, the *Powerful* was to start on a preliminary trial, but this was postponed on account of stormy weather and the fouling of one the propellers by a rope. On the next day a run was made, the engines being worked up to 96 revolutions, at which speed the I.H.P. was 16,100. On the 25th, the first official 30 hours' trial was begun, but, on account of the violence of the storm, was stopped at the end of about 10 hours. The mean draught of the vessel was 27 feet ½ inch, the displacement being 14,200 tons. The means for eight hours were: Steam pressure, at boilers, 223, and at engines, 160 pounds per square inch; vacuum, mean, 26.65 inches; revolutions, starboard, 63.5, port, 63.03; I.H.P., starboard, 2,413, port, 2,540, total, 4,953; coal per I.H.P. per hour, 2.2 pounds; speed, by log, 13.4 knots. Air pressure at nozzles over grate bars, from 5 to 8 pounds above the atmosphere; vacuum in smoke boxes, 22 inches.

16 out of the 48 boilers and 34 out of the 87 auxiliary engines were used. The coal burned was Nixon's hand-picked Navigation. In the first three hours the amount was 2.3 pounds per I.H.P., which was slowly reduced to 2.1 pounds during the last hour. No deduction was made for the fan engines, distillers,

electric lighting, steering, fire and bilge pumps, air and circulating pumps for auxiliary condenser, and the rain tank engine.

The 30 hour's trial was resumed on September 28 at 9.45 A. M., and finished on September 29 at 3.45 P. M., four runs over the mile being made during this time. It may be well to say that the coal used was weighed, and the amount expended for 24 consecutive hours, selected by the Trial Board, was regarded as the mean consumption. The first and last three hours were omitted, giving the average for the remaining 24 hours, 2.09 pounds per I.H.P. The contractors were required not to exceed the guarantee—5,000 I.H.P.—by more than 5 per cent. The same number of boilers and engines were used as on September 25. Only one smoke pipe was in use.

The means of the 30 hours' trial on September 28 and 29 are as follows:

Draught of water, forward, feet and inches	27–3	
aft, feet and inches	27–2	
Steam pressure in boilers, pounds per square inch	225.2	
	Starboard.	*Port.*
Vacuum, inches	26.8	27
Revolutions per minute	67.4	67
Mean pressure in cylinders, H.P., pounds	35.6	37.6
I.P., pounds	11	11.8
L.P., forward, pounds	5.6	6
L.P., aft, pounds	5.6	5.3
Indicated horse power, H.P.	927	972
I.P.	694	742
L.P., forward	417	446
L.P., aft	417	393
Total indicated horse power	2,455	2,553
Collective indicated horse power	5,008	
Speed, by log, knots	14	
Coal per I.H.P. per hour, pounds	2.07	

The means for the four runs over the mile were: revolutions per minute, starboard, 68.8, port, 67.9, average, 68.35; speed (third mean), 14.347 knots; I.H.P., starboard, 2,597, port, 2,604, total, 5,201; slip of propellers, 11 per cent.

The coal consumed per square foot of grate surface was 14.2 pounds, the air pressure at nozzles varying from 8 to 10 pounds

above the atmosphere. The auxiliary machinery probably required the steam generated by 1½ out of the 16 boilers in use.

The rate of consumption during the first of the 30 hours was 2.1 pounds per I.H.P. per hour, the minimum was 1.77 and the maximum, 2.6 pounds, the latter result being due to the cleaning of the fires and tubes of 10 out of 16 boilers simultaneously. During eight hours it was under 2 pounds. The consumption generally ran about 2 pounds to 2.1 pounds; greater variations were due principally to the fact that the coal was weighed out in advance of the requirements of the stokers, and at times accumulated on the plates.

The general result is in part due to the strict observance of method in the stokehold. First, regularity of stoking was insisted upon. Each fire was replenished every eighth minute, so that with four double-fire boilers in a group, each boiler was fired in part every four minutes. This regularity will easily be insured in service; in the meantime cardboard dials are placed around clocks in each stokehold indicating when the fires should be supplied with fuel, but in time these may be dispensed with. Again, only a thin sprinkling of coal was applied at each firing, so that the fire was kept about 6 inches thick and in an incandescent state. The ashes were drawn from the pits and most of them used at the back and wings, as required, and instead of the ordinary raking only the clinker was removed by a two-prong rake. The ratio of ash or clinker was, therefore, small. It may be urged that these precautions necessitate an increase in the staff and involve trouble; but obviously every advantage gained —in respect of rapid steam generation, high pressures and reliability—must involve some extra labor, while the insistance of methodical and careful stoking is of itself commendable for all boilers. It is too often done at hap-hazard. Another element in the economy was the minimizing of the circulating water in the condenser. The small volume of steam for the low power with a large discharge of water tended to reduce the temperature of the feed. In this instance a readjustment—almost closing the outlet valve—gave an increase from 65 to 94 degrees.

The Belleville boilers worked very well, the automatic feed

apparatus giving no trouble whatever. Of course, care was taken every four hours or so to remove grit or dust from the top of the spindles, &c. The strokes of the feed pumps were counted and the volume of feed estimated. It worked out to between 18 and 20 pounds per I.H.P., but the result can only be approximate, since there are many elements of doubt—the efficiency of the pumps, the leakage of the plungers, &c., and the power of the auxiliary machinery; but the evaporative efficiency of the boilers was practically 9 pounds of water from 90 degrees. Nor were the boilers forced. The ashpit doors were very seldom open; they were more frequently closed.

The engines were linked up to give a cut off at .42 of the stroke. The revolutions varied in the case of the starboard engine from 63 to 69, and, in the case of the port, from 64 to 68, according to the wind and tide resistance, while the total power varied from 4,355 to 5,314. The low power mentioned is fully accounted for by the necessity of adjusting the reducing valve and the probability of the engine throttle valve having closed a little by vibration unobserved. The machinery worked very steadily, and there was no hitch.

[NOTE.—Commenting on the above coal consumption trial, a correspondent in "Engineering" compares the result of 2.07 pounds per I.H.P. per hour, notwithstanding that "the combustion was carried on with the greatest possible care," with the 1.6 and 1.7 pounds obtained with cylindrical boilers on the similar trials of the *Venus* and *Minerva*. These trials, and the more recent one of the *Diana*, are published below.—EDIT.]

Between September 29 and October 13, when the trial at 18,000 I.H.P. was commenced, two preliminary runs were made. The force of the wind during these trials was from 8 to 9. There was no pitching of the vessel and the rolling seldom exceeded 17 degrees, and then only at turning. On one of the trials, before coaling, 1,000 tons of water were pumped into the double bottoms to bring the draught to 27 feet, but even then each roll was performed in about 7 seconds, while with the metacentric height normal, the period would be about 9 seconds. The vibration attained its maximum when the engines were running at from

92 to 95 revolutions, the period of vertical oscillation of the ship synchronizing then with the vibratory motion of the engines; but even then it was not abnormal, certainly under ¼ inch towards the center of the nodes. Before the engines reached 100 revolutions the vibration had practically ceased, and at 103 revolutions, the ship was steady, excepting such slight motion, principally lateral, as was set up when one engine revolved at a different speed from the other.

Elements contributing to this result are the arrangement and proportions of the engines. Without the use of extra weights the moving parts are well, although not absolutely, balanced. As the two L.P. cranks are opposite each other, there is perfect equality. The H.P. and I.P. cranks are opposite; the only variation in weight is owing to the different sizes of cylinders. The H.P. piston weighs 1,901, and the I.P., 3,479 pounds, a small difference when the total weights of the moving parts are considered. The L.P. pistons weigh 3,752 pounds each.

Another point which may be referred to here is the arrangement made for the vibratory working of the upper structure. With the armored deck this "working" cannot be appreciated below; but the light boat deck, extending from the forecastle head to the poop, and for only part of the width of the ship, is only of ¼-inch steel, supported on the frames and shell plating, and on stanchions. At two points in its length, gaps or expansion joints 2 inches wide have been left athwartships. These divide the deck from bow to stern into three equal parts, and as they adjoin baggage ports on the side plating, it became easy to continue the gap down the shell plating to the upper deck. The "play" at the expansion joint coincided with the vibration of the ship, and there was no strain involved by "working."

The official trial began on Tuesday, October 13, and ended 30 hours afterwards, or on October 14.

The mean results were as follows.

Draught, forward, feet and inches	27–4
aft, feet and inches	27–5
Steam pressure in boiler, pounds per square inch	232
Displacement, tons, about	14,200

	Starboard.	*Port.*
Vacuum, inches	26.7	26.5
Revolutions per minute	103.62	101.98
Mean pressure in cylinders, H.P.	83.44	87.66
I.P.	27.19	27.46
L.P., forward	13.96	14.34
L.P., aft	14.05	14.03
I.H.P., H.P.	3,337	3,447
I.P.	2,696	2,615
L.P., forward	1,552	1,609
L.P., aft	1,603	1,574
total	9,188	9,245
collective	18,433	
Coal per I.H.P., mean for 30 hours	1.838	

All boilers, or 48, were used. From the total coal burned on the trial, 8 tons were deducted because there had been made during the trial a large quantity of fresh water, far in excess of the usual requirements of the boilers or of the ship. This deduction made a fractional difference in the mean result, 1.838 against 1.87 pounds per I.H.P. per hour. During the first hour of the run the coal consumption was at the rate of 1.8 pounds per I.H.P. per hour, and throughout the whole run it remained remarkably uniform. Twice only did it exceed 2 pounds, and then it was 2.08 and 2.02 pounds, while twice the consumption was below 1.7 and seven times below 1.8 pounds.

The stokers worked with as little supervision as in service. Clinker was removed every four or five hours; ashes were used for the wings, as in the former trial. The boilers worked well, and with a sufficient supply of compressed air there was an absence of smoke and of flame from the chimney. Only once during the night, when the compressed air was low, was there even a suggestion of flame from the chimney. The air blowers registered from 10 pounds to 12 pounds. The vacuum in the smoke boxes averaged .4 inch, varying from .37 inch to .44 inch. The ventilating fans were run at 160 to 180 revolutions per minute, but the ashpit doors were seldom further open than the first notch (about 2 inches). Only one circulating pump was in use, with the inlet full open, and the mean temperature of the feed was about 100 degrees. The consumption of coal was 16.1 pounds per square foot of grate area.

The total quantity of coal burned was 372 tons, which includes the consumption for 59 auxiliary machines. The Normandy distillers were worked so that the amount of fresh water in the ship was the same at the end as at the beginning of the trial.

"The number of merchant steamers capable of such an economical performance as was shown by this trial is very limited. Moreover, we take several warship performances, chosen at random, and for an entirely different purpose. The *Magnificent* working at cruising power (three-fifths natural draft) consumed 1.69 pounds per unit; the *Majestic*, at five-sixths full power, 2.07 pounds; the *Prince George*, at three-fifths natural draft, 1.83 pounds; the *Talbot*, at three-fifths natural draft, 1.84 pounds; the *Victorious*, at three-fifths natural draft, 1.6 pounds. These figures seem to indicate that it would be well to suspend condemnation of the Belleville boiler on the score of the economy, at all events until there is fuller experience."

The engines worked well. They were linked up to .6. The highest power developed was 19,000, with the starboard engine running 104.2, and the port engine 102.6 revolutions, the corresponding powers being 9,398 and 9,602. The lowest record was 17,266, when the revolutions starboard were 102 and port 99.4. This unusually low record was due to a change of course and an adverse direction of wind, while the cowls were neglected for a short time.

On Monday, and in the early morning of the day on which the 18,000 I.H.P. trial commenced, six runs were made over the course between Rame Head and Dodman Point, the distance being between 22.5 and 23 nautical miles. The results for the first three runs were: I.H.P., 17,991; revolutions per minute, 102; speed, 20.63 knots. For the second set, the results were: I.H.P., 18,678; revolutions per minute, 103.2; speed, 20.96. The slip of propellers was from 13.5 to 14 per cent. The ratio of surface area of propellers to disc area is about 1 to 30.

On October 17, a preliminary high speed run was made, the engines working up to 110 revolutions, and indicating 24,800 horse power.

Some slight adjustments, usual in the case of all such high

speed machinery, were made, but are not worth detailing. It may be interesting to state, however, that on the suction side of the feed pumps air vessels have been fitted.

The *Powerful* went out on October 21, to run at full power (25,000 I.H.P.). For two hours the ship steamed well, everything working satisfactorily. The coal was weighed during this period, and worked out to just over 2 pounds per I.H.P. per hour; but the weighing of the coal had to be discontinued, as it somewhat shortened the supply. At this time also the gauge pipe of one of the boilers—a solid-drawn copper pipe—gave way, so that there was one boiler short for the greater part of the run. The mean power for four hours was 24,700 I.H.P. The fires clinkered heavily after the first hour, and in some cases were kept rather too thick for this type of boiler under natural draft. This probably accounted for some heating of the up-takes much in excess of that observed on previous trials. The maximum temperature recorded between the uptake and casing was 180 degrees. The specifications provided that the uptakes, and such bulkheads and passages as are in close proximity to the uptakes, should be lagged with silicate cotton, and stated that on the full-power trial it would be determined where and to what extent this lagging would be adopted. This is the usual procedure. The *Powerful* was taken into the harbor on Thursday forenoon, October 22, and this lagging will now be put on before the ship again goes on her full power official trial. Fires and tubes will be cleaned, and slight adjustments to eccentric straps made. The *Powerful* will probably go out in the course of the next fortnight.

Amphitrite, Argonaut, Ariadne and **Spartiate.**—These four first class cruisers of 11,000 tons displacement, to be built under the programme for 1896–'97 (page 377 of the May number), have just been ordered. The following particulars are from "Engineering," "Broad Arrow," and "The Times," London.

The *Argonaut* will be built and engined by Messrs. J. and G. Thomson, Clydebank; the *Ariadne* by the Fairfield Company, Glasgow; the *Amphitrite* by the Naval Construction and Armaments Company, Barrow-in-Furness, and the *Spartiate* at the Pembroke dockyard. The engines of the last named are to be

made by Messrs. Maudslay, Sons & Field, London. The vessels are to be of the same type as the *Niobe*, and are to be 435 feet long, with a beam of 69 feet and a depth of 39 feet 9 inches. They will have a forecastle and a boat deck, but no poop. They will be built of steel and sheathed, the stem and stern post being of bronze. At 25 feet 3 inches draught, the displacement will be 11,000 tons. The armament will consist of sixteen 6-inch quick firers, twelve 12-pounders of 12 hundred weight, two 12-pounders of 8 hundred weight, three 3-pounders, eight .45-inch Maxims, two submerged torpedo tubes and one above water at the stern. The protective deck will vary from 2.5 inches on the crown to 4 inches on the slope, and have a rise of 10 feet. There will be coal bunkers below and above the deck, along the sides of the machinery and boiler compartments. There will be two sets of four-cylinder, triple expansion engines, with cylinders 34, 55½ and (2) 64 inches in diameter, with a 48-inch stroke. Instead of having the H.P. cylinder forward, with the I.P. next and the two L.P. aft, it is proposed to put one L.P. cylinder at each end, with the view of economizing space. The engines will run faster than those in the class now building—120 instead of 110 revolutions—so that the power will be increased from 16,500 to 18,000 I.H.P., giving 20.75 instead of 20.25 knots. The Belleville boilers will be slightly different. Over the ordinary series of elements there will be a corresponding number of elements, consisting of smaller tubes, 2¾ inches in diameter, forming economizers, and between the two, a combustion chamber into which air will be injected by nozzles. This has been adopted to improve the circulation, and, consequently, the economy. There will be 30 boilers in all, 18 of them with eight large and eight small elements, and 12 with seven elements. The total heating surface will be 47,880 square feet, of which 15,505 square feet will be in the feed heating tubes. The total grate surface will be 1,390 square feet. The steam pressure at the boilers will be 300, and at the engines 250 pounds per square inch. The total weight of machinery will be about 1,525 tons. The contract price for each cruiser is said to be about $2,225,000.

Diadem.—This first class sheathed cruiser, a description of

which was given on pages 813 and 814 of Volume VII, was successfully launched by the Fairfield Shipbuilding and Engineering Co., Govan, on October 21. The following description of the machinery and armament, and some of the hull data, are from "Engineering," London.

The length over all is 462.5, and between perpendiculars, 435 feet; beam, 69 feet, and displacement, 11,000 tons. The keel was laid in January, 1896. The hull is of steel, sheathed with teak 4 to 4.5 inches thick, and coppered. Bilge keels are fitted. The double bottom is 3.5 feet deep in the center, and 2.5 feet at the sides. The hull plates are 22.5 pounds to the square foot. The machinery space occupies 188 feet of the total length of the vessel. The stem, sternpost and shaft brackets are made of phosphor bronze, as usual in sheathed vessels. The rudder, which is of the balanced form, is a casting of phosphor bronze, plated with naval brass, and is controlled by Harfield's patent compensating gear, in addition to the steam steering engines, all of which are arranged below the protective deck. There are two signal masts, fitted with steadying sails. The coal capacity at normal draught is 1,000 tons, but provision has been made for carrying about twice that quantity if occasion should require.

The armament differs somewhat from that originally intended, and will consist of four 6-inch quick firing guns, with shields, on the upper deck; twelve 6-inch quick firing guns, fitted in armor casemates of harveyized steel, on main deck; sixteen 12-pounders; a large number of smaller and machine guns; two torpedo tubes discharging below water forward, and one above water through the stern.

There are two sets of vertical, four-cylinder, triple expansion engines on each shaft, with cylinders 34, 55.5 and (2) 64 inches in diameter, the stroke being 48 inches. The H.P. and I.P. cylinders are each fitted with one piston valve, and each of the L.P. cylinders with a double ported slide valve with relieving ring at the back. The ordinary link motion valve gear is used. The cylinder covers, pistons and bed plates are of cast steel. The cylinders are supported at the backs by cast iron columns and at the front, by forged steel columns.

The crank shafts are solid double-throw, and made interchangeable, and, together with the thrust and propeller shafts, are of steel and bored hollow. Each of the propellers has a boss of gun-metal, fitted with three adjustable blades of manganese bronze, and constructed to work inwards.

The main condensers are placed at the back of the engines, constructed of cast brass, and of oval form, and fitted with brass tubes $\frac{5}{8}$ inch in diameter, the condensing water being supplied by four centrifugal pumps of gun metal, each fitted with independent engines. The auxiliary condensers, one in each engine room, are of cast brass, with separate centrifugal pumps and engines. These condensers take all the exhausts from the small engines about the ship.

The 30 Belleville boilers will be in four compartments, each compartment having a separate smoke pipe.

The complement will consist of about 700 officers and men.

Venus.—This second class cruiser, a description of which was given on page 815, Volume VII, has completed satisfactory official trials.

She was delivered at the Chatham dockyard by the builders, the Fairfield Shipbuilding Co., Govan, about the middle of June, considerably ahead of the contract date.

The following additional particulars are from "Engineering" and the "Times," London.

The double bottom extends the whole length of the vessel. Side compartments under the protective deck, for the whole length of the boiler space, are used as coal bunkers; cross bunkers are arranged at the ends of the two boiler compartments, so that each set of four boilers is surrounded by coal. Reserve bunkers are fitted on the protective deck over the machinery space. The number of water tight doors has been reduced to a minimum, and all doors can be worked from the main deck and from below. The sternpost, struts and stem are of phosphor bronze. The rudder, also of phosphor bronze, is of the balanced type, and controlled by Harfield's compensating gear below the protective deck.

The teak sheathing is 3.5 inches thick at its thinnest part. The bilge keels extend for about one-half the vessel's length.

In addition to the conning tower forward, there is a director tower aft, for operating the torpedoes. There are two bridges, on which are three search lights. The magazines and shell rooms are of ample capacity, and conveniently situated for the rapid fire guns, special gear for manipulating the ammunition being fitted. The coal capacity is nominally 550 tons, but about 1,000 tons can be carried. There are two masts with military tops, and rigged with fore-and-aft steadying sails.

The armament consists of five 6-inch, eight 4.7-inch and eight 12-pounder rapid fire guns, all protected by extra heavy shields. There is besides a number of machine guns.

The complement of officers and men will be 450.

The twin screw, inverted, triple expansion engines are in separate compartments. The cylinders are 33, 49 and 74 inches in diameter, the stroke being 39 inches. The cylinders have cast iron liners and cast steel covers, and are steam jacketed. The main valves are worked by double eccentric and link motion gear, balance cylinders and compensating rings being fitted to the flat slide valves. The reversing engines are of the all-round type, with worm and wheel gear, and all the reversing levers have a slot with an adjusting screw to allow of variable expansion in the cylinders.

The back columns are of the split type, of cast iron, and fitted with separate guide faces. The front columns are of forged steel, the engines being arranged with the starting platforms amidships. The main condensers are placed in the wings, and are of cast brass, with solid drawn brass tubes, the water passing through the tubes. There are two large centrifugal circulating pumps of cast brass, each worked by an independent engine, one in each engine room, and arranged with a cross connection, so that either or both condensers can be supplied with water from either pump. The main feed, auxiliary feed, bilge and fire engines are all independent, and are arranged on the center line bulkhead, steam being supplied by a special line of piping used for auxiliary purposes only. Auxiliary condensers of cast brass,

similar to the main condensers, are fitted. A feed water filter is fitted between the air pump and feed tank.

The crank, thrust and propeller shafting is hollow, of Whitworth's fluid compressed steel. The crank pins are fitted with centrifugal lubricators. The propellers are three-bladed, the bosses and blades being of gun metal. The main steam pipes are so arranged that steam from the boilers in either boiler room can be used for the engines in either or both engine rooms.

The following are the results of the trials.

Four hours' forced draft trial: Steam pressure, 145 pounds per square inch; vacuum, starboard, 26.1, port, 25.65 inches; revolutions per minute, starboard, 145.9, port, 148.4, mean for both engines, 147.15; I.H.P., starboard, 4,913, port, 4,831, collective, 9,744; speed, 20.18 knots.

Eight hours' natural draft trial: Steam pressure, 150 pounds per square inch, never varying more than 5 pounds during the trial; vacuum, starboard, 25.6, port, 26.2 inches; revolutions per minute, starboard, 136.6, port, 137.2, mean for both engines, 136.9; I.H.P., starboard, 4,068, port, 4,136, collective, 8,204; speed, 19.25 knots.

Thirty hours' coal consumption trial: Vacuum, 26.85 inches; revolutions per minute, 118.05; collective I.H.P., 4,876; speed, 16.8 knots; coal per I.H.P. per hour, 1.6 pounds. This coal consumption is less than all other ships of the same class that have yet been tried.

Of the six vessels ordered from private yards in May, 1894, the *Juno*, *Doris*, *Dido*, *Isis*, *Venus* and *Diana*, the *Venus* is the first to be delivered, and the sister ship *Diana*, the trials of which are described below, the second.

Minerva.—Another of the dockyard-built second class cruisers of the *Talbot* class has successfully completed her trials. This vessel was floated out of the dock at the Chatham yard on September 23, 1895, the keel having been laid on December 4, 1893.

Preliminary trials held in June were satisfactory, so far as the working of the machinery was concerned. At the inspection afterwards a serious crack was discovered in one cylinder, be-

sides defects in boilers. A new cylinder was put in, the boiler defects corrected, and on September 23 a satisfactory preliminary trial of four hours' duration was made.

On September 24 the eight hours' natural draft trial was made, with the following results.

Steam pressure, 151 pounds per square inch; air pressure, .25 inch; I.H.P., 8,221; speed, by patent log, 19.6 knots, or 1.1 knots over the designed speed.

The four hours' trial under forced draft was made a few days afterwards, with the following results.

Draught of vessel forward, 19 feet, aft, 21.25 feet; steam pressure, 152 pounds per square inch; vacuum, starboard, 27.5, port, 27.7 inches; revolutions per minute, starboard, 137.7, port, 137.5; I.H.P., starboard, 4,968, port, 4,923, total, 9,891; speed, by log, 20.34; air pressure, 1.02 inches. The I.H.P. developed and the speed attained are the greatest for the vessels of this class.

The thirty hours' coal consumption trial was finished September 30, and resulted as follows. Draught of water, forward, 18 feet 8 inches, aft, 21 feet, mean, 19 feet 10 inches; steam pressure, 146 pounds per square inch; vacuum, starboard, 28.5, port, 28.4 inches; revolutions per minute, starboard, 111.5, port, 111.7; I.H.P., starboard, 2,437, port, 2,482, total, 4,919; speed by log, 17.52 knots; coal per I.H.P. per hour, 1.7 pounds.

Diana.—This second class, sheathed cruiser, built by the Fairfield Shipbuilding and Engineering Co., Govan, and described on page 170 of the February number, has completed satisfactory official trials. The vessel was delivered by the contractors at the Sheerness dockyard on September 13, and there prepared for the trials.

The eight hours' trial under natural draft was made on October 17. The draught of vessel was 19.5 feet forward and 21.5 feet aft; steam pressure in boilers, 145 pounds per square inch; vacuum, starboard, 26.4, port, 26.9 inches; revolutions per minute, starboard, 136.8, port, 135.8; I.H.P., starboard, 4,206, port, 4,046, total, 8,252; speed, 19.72 knots; consumption of coal per I.H.P. per hour, 2.25 pounds; air pressure, .46 inch.

On October 20, the four hours' trial under forced draft was completed. The contract I.H.P. of 9,600 was exceeded by 275. The speed was 20.16 knots.

The thirty hours' coal consumption trial was finished on October 24.

Draught of vessel, forward, 18.75 feet; aft, 21.5 feet; steam pressure in boilers, 142 pounds per square inch; vacuum, starboard, 26.6, port, 26.9 inches; revolutions per minute, starboard, 116.2, port, 116; I.H.P., starboard, 2,459, port, 2,457, total, 4,916; coal per I.H.P. per hour, 1.47 pounds; speed, 17.24 knots.

Pactolus and **Pomone.**—By the naming of the *Pomone*, the list of the six third class cruisers authorized this year has been completed. The general data of these vessels were given on page 391 of the May number. The *Pomone* will be built at the Sheerness dockyard. The contract for the machinery of the *Pactolus* has been given to Messrs. John Penn & Sons, Greenwich. The machinery of the *Pomone* will be built at the Devonport dockyard.

The engines will be of the vertical, inverted, triple expansion type, with cylinders 20.5, 33 and 54 inches in diameter, the stroke being 27 inches. These engines will be the same as those for the earlier cruisers, *Pelorus* and *Proserpine*. There will be piston valves on the H.P. and I.P. cylinders. There will be eight Blechynden water tube boilers on the *Pactolus*, and Thornycroft water tube boilers on the *Pomone*, the working pressure to be 300 pounds per square inch at the boilers, reduced to 250 pounds at the engines. The total heating surface will be about 17,504, and the grate surface about 351 square feet.

Working at 220 revolutions per minute, the engines are expected to develop 7,000 I.H.P., and give a speed of 20 knots.

According to the "Naval and Military Record," London, "a new feature is to be adopted in the construction of the machinery, it having been decided to electroplate the steam pipes internally and galvanize them externally, this precaution being taken to prevent corrosion. Recently there have been several machinery breakdowns in consequence of defective steam pipes, which the naval engineers have stated would not have occurred but for

corrosion, and the Admiralty intend in future to have the steam pipes of all new machinery subjected to the electro-plating and galvanizing process."

Spanker.—The trials of this torpedo gunboat, with the new DuTemple boilers, were given on page 392 of the May number. The following data of the new boilers are from a paper by Mr. G. Malfatti, Engineer, Italian Navy, published in the "Rivista Marittima," Rome.

There are four boilers, designed for a working pressure of 200 pounds per square inch. They have a total grate surface of 172.2, and a total heating surface of 8,052 square feet, the ratio of heating to grate surface being 46.75 to 1. The weight of the four boilers complete, without water, is 47.24 tons; weight of water, 11.81 tons; total weight, 59.05 tons.

Janus, Lightning and **Porcupine.**—A description of these torpedo boat destroyers, built by the Palmer Shipbuilding and Iron Company, Jarrow-on-Tyne, with some results of their trials, were given on page 425 of Volume VII and pages 173 and 398 of the current volume. The following additional description of the machinery, and a comparative table of the trials, is from "Engineering," London:

The vessels are built of steel rolled at the steel works department of the company, and the galvanizing was also carried out in the same section of the works by electro deposition. It may be stated that Palmer's works have also supplied the material of which other destroyers have been built by different firms. The vessels and their machinery were both designed by the company in accordance with the conditions laid down by the Admiralty as to speed, armament, load to be carried, &c.

The trials were made off the mouth of the Thames, and during that of the *Lightning* there was a strong wind and heavy sea. In spite of this, at one time the vessel was going through the water at the rate of over 29 knots. The machinery and boilers were reported to have worked satisfactorily, the latter giving an ample supply of steam, the air pressure being, as will be seen by the table, by no means high.

The following are the results of the official trials of these boats:

	Janus.		*Lightning.*		*Porcupine.*	
Date of trial	August 6, 1895.		November 5, 1895.		February 19, 1896.	
	Six runs on measured mile.	Three hours.	Six runs on measured mile.	Three hours.	Six runs on measured mile.	Three hours.
Revolutions per minute... Starboard	367.67	365.8	378.2	371.2	363.9	360.8
Revolutions per minute... Port	370.72	368.5	375 9	365.4	365.2	362.0
Revolutions per minute... Mean	369.19	367.1	377.1	368.3	364.5	361.4
Steam pressure, pounds per square inch	192	193	187	192	198	197
Vacuum, inches	26.25	26.1	26.7	26.5	26.7	26.7
Indicated horse power	3,750	3,789	3,925	4,000	3,862	3,866
Air pressure, inches	2.6	2.48	2.26	2.48	2.07	2.22
Speed, knots	27.728	27.805	28.274	27.944	27.989	27 915

The consumption of fuel on the various boats ranged from 14 to 15 tons of coal for the three hours' trials. These boats are stated to be very free from vibration, a fact which is certainly true of the *Porcupine*, and doubtless of the other vessels. It may be noted that less than 2½ inches of air pressure was sufficient, although, owing to the arrangement of the tubes, &c., in the type of boiler used, more resistance is offered to the passage of gases than is the case in some of the straight tube boilers.

It is worth noting that there was no alteration from the original design of the machinery, the propellers of the three vessels being identical and similar to the first pair tried.

The engines have cylinders 18, 27.5 and 42 inches in diameter, the stroke being 18 inches. The bedplate is formed by two rolled steel angle bars with five cross girders of cast steel between, to carry the bearings of the crankshaft in the usual way. The bearings are of white metal, and have forged steel keeps. The turned columns are of steel, and are rigidly braced transversely and longitudinally. The cylinders are cast with solid bottoms and are bolted together; the steam connections are formed by copper pipes. Piston valves are fitted to all cylinders, the high and intermediate pressure valves being of cast iron, whilst the low pressure valve is of aluminum bronze. The cylinder

covers are of cast steel. The pistons are of forged steel and are machined all over. They are fitted with Ramsbottom rings. The piston rods and connecting rods are of steel, and are forged hollow, whilst all unnecessary metal about the heads is machined off for lightness. The piston rod shoe is solid with the rod, and is lined with white metal on the ahead surface for working on the cast iron guide bars; the latter being hollow and having the water service usual with engines of this nature. The guide bars are attached at the top to brackets depending from the cylinders, and at the bottom ends there are horizontal forged bars between the columns to act as supports. The three-throw crankshaft is a hollow steel forging, and has the large bearing surfaces necessary with quick running engines of the type. Balance weights are attached to the crank webs. The valve gear is of the ordinary double-bar type, the forgings being of steel. The eccentric straps are of gun metal, lined with white metal. The reversing shafts are hollow steel forgings in two pieces. One of the levers is forged on the coupling. The reversing gear is for hand power only, but has worked well. The time on trial, from full speed ahead to astern was seven seconds. All handles for controlling the engines are at the forward end, across which a starting platform has been arranged.

The condensers are cylindrical, of sheet brass riveted together. The centrifugal circulating pumps are at their forward ends, the inlet and outlet pipes being led in as direct a manner as possible. There is a slight lip projecting beyond the skin of the vessel on the inlet, and it has been found that a vacuum of 27 inches can be maintained without the circulating pumps when steaming at 27 knots.

The air pump is driven by a double crank forward of the end of the main crank shaft, the pump being secured to the bedplate girders, which are prolonged for the purpose. At the after end of the engines, the thrust block is attached in a similar manner, a disposition which makes the engines self-contained. The arrangement of the engines is such that there is a passage down the center between the engines, and also a way at each wing between the engines and the condensers. There is also a passage

at the back, and with the starting platform at the forward part, free access is given all round the engines.

Lubrication is well provided for, as is necessary in quick-running engines of this type. The oil boxes are fed from tanks high up in the engine room, and are fitted with sight feeds underneath. The usual water service for cooling is attached, that for the thrust blocks having branch pipes leading to each horse shoe. The thrust shaft and stern shaft are of hollow forged steel. The stern tubes are formed by steel tubes built into the ship and fitted with cast steel bushes at each end, these bushes being lined with white metal. In both the stern tubes and outer brackets, the bare steel shafts work on white metal without any gun-metal lining. The propellers are three-bladed, of aluminum bronze, the blades and boss being cast in one. The propellers are entirely below the bottom of the vessel, so as to insure as far as possible that they should work in water unmixed with air.

Special care has been taken in the design of these engines to avoid vibration, both by apportioning the balance weights and by staying the engines within themselves. The two sets are also stayed together and to the hull at the tops of the cylinders. In addition, the two vertical plates forming the outer girders of the engine seating are continued through the boiler room to afford supports for the boilers, and thus constitute two plate girders from the after end of the engine room to the forward end of the boiler room. Judging by the trial of one of these vessels which we attended, the efforts made in this direction have been attended with success. We understand that an experiment, made by running one of these engines without balance weights, indicated that the freedom from vibration was principally due to the presence of these weights, although doubtless the staying of the engines and the stiffness of the hull structure greatly assisted towards the desired end.

The air compressing machinery, together with evaporating and distilling machinery for feed make-up and the supply of fresh water for drinking purposes, are in the engine rooms. A separate duplex pump is fitted for pumping out the bilges. The steam steering engine is placed on the after engine room bulk-

head. The feed tanks in the engine room are fitted with a simple form of grease filter. A small air pump is attached to each centrifugal pumping engine to keep the main condensers free from water when the main engines are stopped.

The water tube boilers of these vessels are designed by Mr. J. W. Reed, the chief engineer to Messrs. Palmer's, and have a working pressure of 210 pounds to the square inch. They have three drums, two in the wings and one above, with the generating tubes connecting them, and there are two large external downcast pipes at each end, so arranged as to form a substantial part of the framework of the boiler. The chief object in fitting the downcomers is to provide ample circulation of water, so as to give a good supply to the bottom ends of the tubes. A manhole is provided in the ends of the upper chamber to give free access to the interior. The tubes are bent so that the rows on each side nearest the fire meet in the middle, and thus form a crown to the furnace, protecting the top barrel in a manner similar to that adopted in the Thornycroft boiler. There is also a protecting shield of fire tiles or other refractory material placed above the point where these two rows branch off from each other before entering the top drum.

The rows of tubes nearest the fire are not only curved to the contour necessary to form the furnace, but are bent in the manner of a flattened zigzag or wave form where they are exposed to the full force of the radiant heat of the fire.

The method of attaching the tubes to the curved tube plates which form the top drum and wing chambers, respectively, is designed with a view to enable the joints to accommodate themselves to any slight deviation of the tubes from the correct line. A semi-spherical washer or ferrule is secured to the tube on the inner side of the tube plate, whilst there is a corresponding orifice in the tube plate to take the washer. When the nuts on the tubes are screwed up, the spherical surfaces are drawn into close contact with one another to form an effective joint. Mr. Reed has also designed another form of joint which is a further extension of this principle, and which has a second spherical fitting, the nut on the tube outside of the tube plate being dished to fit

over a washer with a correspondingly formed back. This would appear to carry out his intention more fully, but the arrangement described is that adopted in the boilers in question, and has been found to answer the purpose for which it was designed. In these tubes, we are informed, it was rather difficult to get the exact bend so as to make them lead fair within the holes, but by the spherical joint, the tube could be drawn up and the joint made secure, even if the tube end were not fairly in line. The face where the nut goes on is intended to be at right angles to the axis of the intended central line, but owing to the bottom chambers being made of bent boiler plates, it was found that accuracy was not always obtained in actual work.

The tubes are of solid drawn steel, galvanized externally by electro deposition. By means of the particular joint described they are very readily placed in the boiler and secured. In the boilers of these vessels there are 30,000 of these joints, and we understand that not one has been known to leak. Baffle plates are placed at the sides and over the tubes, to assist in deflecting the gases. The casing or smoke jacket surrounding the furnace and tubes is double, with an air space between, and is lined on the fire side with an asbestos composition. The arrangement is satisfactory, radiation being reduced to a very low point, it being possible to place one's hand on the boiler front when working with heavy forced draft. The part immediately in contact with the fire is lined with fire quarls. Three fire doors are provided, which close and fasten automatically. The system of supplying air to the furnaces through non-return doors is adopted in order to prevent flame passing into the stokehold in case of a burst tube. We are informed that the twelve boilers for these three vessels have been many times under steam, in some cases, for experimental purposes, being driven with an air pressure equal to $5\frac{1}{2}$ inches, but there has not been an accident or leaky joint.

The main steam pipes are solid drawn steel tubes with riveted flanges, a separate pipe being led from each boiler room to the engines. A complete system of steam pipes is also fitted for supplying the auxiliary machinery, and there is a separate system of exhaust pipes from the auxiliary engines to both condensers.

The boilers are arranged two in a compartment. The forward and after boilers have each a separate funnel, the two center ones having one large funnel common to both. Two single-breasted fans are fitted in the stokehold for forced draft. There is one main feed pump to each boiler, controlled by an automatic feed regulator designed by Mr. Reed. In the stokehold there is also a third duplex pump for auxiliary feed. These pumps draw from two feed tanks in the engine room, the main air pumps delivering into them.

The first boiler made was tested on shore, and an evaporation at the rate of 28,300 pounds of water per hour was recorded. On an evaporation trial of six hours' duration, made with one of the boilers of the *Lightning* when fitted on board, an evaporation of 12 pounds of water per pound of Welsh coal was recorded. The boilers are reported to be capable of supplying more steam than the engines can take. The *Porcupine* was taken out to sea with only half her boilers in action to see what speed could be obtained under these conditions. The two boilers supplied steam for about 2,800 I.H.P., or 1,400 horse power each, with less than 4 inches of draft, the mean speed of the vessel being 23.79 knots. One boiler weighs 13¼ tons complete, with furnace gear, clothing, mountings, water, &c.

Recruit.—This 30-knot torpedo boat destroyer, one of the four building by Messrs. J. and G. Thomson, Clydebank, was launched on August 22. As stated on page 820 of Volume VII, the displacement is 300 tons and the expected I.H.P. 5,800.

Star and **Whiting.**—These 30-knot torpedo boat destroyers, two of the eight vessels mentioned on pages 401 and 622 of the current volume, building by the Palmer Shipbuilding and Iron Company, Jarrow-on-Tyne, were launched from their Howden yard in August. The following particulars are from "Engineering," London.

Length, 215 feet; beam, 20.75 feet; displacement, about 300 tons. The armament will consist of one 12-pounder on the conning tower, four 6-pounders in broadside and one 6-pounder on a platform aft, all rapid fire guns, and two revolving torpedo tubes on deck.

The twin screw engines are of the triple expansion type. There are four Reed water tube boilers.

Desperate.—The repairs involved by the breakdown, mentioned on page 621 of the August number, having been completed, the official trial was made on September 4. Previous trials (and the new requirements exacted on the trials of these vessels) were described on page 399 of the May number. The following data are from "The Engineer," London.

Mean results of the six runs on the measured mile. Revolutions per minute, starboard, 400.15, port, 406.47, mean, 403.31; speed, 30.428 knots.

The mean speed for the three hours' run was 30.018, for which the corresponding revolutions would be 401.49. The coal consumption per I.H.P. per hour was 2.43 pounds.

A slight defect having made itself apparent towards the end of this trial, a supplementary three hours' trial, under the same conditions as above, was made on September 24, with results entirely satisfactory to the engineer officers responsible for the final acceptance of this vessel. The *Desperate* is the first to be accepted of the twelve 30-knot destroyers ordered last year and mentioned on page 820 of Volume VII.

Sparrowhawk.—This torpedo boat destroyer, built by the Messrs. Laird Bros., Birkenhead, and described on page 820 of Vol. VII, completed a three hours' preliminary trial on July 24, making a speed of 30.25 knots. The displacement on trial is given as 220 tons. The water tube boilers are of a modified Normand type. On September 14, another trial was made in a rough sea, the speed being 30.5 knots. In the beginning of October, the contractors' trials were successfully completed, and on October 16, the official trial was made. The revolutions per minute on the measured mile were 364, the speed being 30.2 knots. The mean speed for the three hours was 30.68 knots. The full contract load was on board.

On October 24, another three hours' trial was made to determine the coal consumption at 30 knots. The mean results for the three hours were as follows. Speed, 30.03 knots; I.H.P., 6,000; coal per I.H.P. per hour, 2.49 pounds of Welsh coal.

Opossum.—While in the hands of the dockyard, whither she had gone to renew damaged propeller blades, caused by striking a rock, defects in her boilers were discovered. After extensive repairs, the vessel started on a three hours' trial on July 27. This trial was interrupted by the breakdown of the P.H.P. engine. Examination at the Devenport dockyard showed that the slide valve and valve chest bonnet were broken and the rod badly distorted.

The results of the official trials, made in February, were given in detail on pages 396 and 397 of the May number.

Spitfire.—This torpedo boat destroyer, launched on June 7, 1895, was delivered to the Government on July 30, 1896, by the contractors, Messrs. Armstrong, Mitchell & Co., Newcastle. The sister vessel, the *Swordfish*, was described on page 618 of the August number. It may be added that the designed displacement of these vessels is 265 tons, and that their armament consists of one 12-pounder and five 6-pounder rapid fire guns and two torpedo tubes.

On August 19, the three hours' official trial was made, the results being as follows. Steam pressure at boilers, 195 pounds per square inch; revolutions per minute, 398; I.H.P., 4,527, or 527 over the contract requirement; speed for the three hours, 27.461 knots, for the six runs over the measured mile, 27.775 knots.

Swordfish.—This destroyer, built by Messrs. Armstrong, Mitchell & Co., Elswick, made two other full power trials in October. On the official trial, detailed on page 619 of the August number, the contract speed of 27 knots was not attained, although the I.H.P. was exceeded. From the following data of the first October trial, it will be seen that similar results were obtained.

Draught, forward, 5 feet 1½ inches, aft, 7.5 feet; speed, 26.702 knots; steam pressure in boilers, 199 pounds per square inch; vacuum, starboard, 24.2, port, 24.4 inches; revolutions per minute, starboard, 398.5, port, 392.7; I.H.P., starboard, 2,208, port, 2,227, total, 4,435; air pressure, 2.66 inches. The weather was bad and the sea rough.

Another trial was made, with the following official results. Steam pressure in boilers, 203 pounds per square inch; vacuum,

23 inches; revolutions, per minute, 399; I.H.P., 4,570; air pressure, 2.3 inches. As will be seen, the contract requirements were exceeded.

Foam.—The third torpedo boat destroyer of the *Desperate* class, building by Messrs. J. I. Thornycroft & Co., was successfully launched on October 8, with all machinery and armament on board.

Avon.—This 30-knot torpedo boat destroyer, building by the Naval Construction and Armaments Company, Barrow, was successfully launched on October 10.

Salmon and **Snapper.**—The following details of the machinery of these 27-knot destroyers, built by the Earle's Shipbuilding and Engineering Co., Hull, are from "Engineering," London. These vessels were accepted in 1895, the official trials of the *Salmon* being given in detail on page 817 of Vol. VII. The trial data of the *Snapper*, published now for the first time, are given below.

The length of these vessels is 200 feet; breadth, 19 feet 6 inches; and displacement in service trim and steam up about 250 tons. They are built entirely of steel of light scantling. The whole of the material used was galvanized by the builders by means of their electro deposition plant. The armament consists of two torpedo tubes, one 12-pounder gun mounted on the top of a conning tower, and five 6-pounders, two of which are mounted on each broadside, and one on a raised platform aft.

The main propelling machinery, also designed by the builders, consists of two sets of triple expansion engines with cylinders 19.25, 28.5 and 43 inches in diameter, the stroke being 18 inches. These engines are somewhat novel in design, especially as regards the framing, the usual turned and polished steel back columns having been replaced by manganese bronze frames designed to carry the piston rod guides. The bed plate is also formed of transverse girders of manganese bronze bolted to fore-and-aft rolled steel girders. The front columns are of polished steel, and are well stayed both fore and aft and athwartships. Manganese bronze has been extensively used in connection with the various parts where cast steel is usually employed, in addition to the bed-

plate and frames above mentioned. This change in design is said to have given most satisfactory results. Piston valves of light design are used throughout, being actuated by double eccentric link motion. The links are of the slot type. The valve gear is controlled by a single cylinder steam reversing engine, in addition to the usual hand reversing gear. The air pumps are placed forward of the main condenser, and are worked by means of a crank driven by a connection with the foremost end of the engine crankshaft, and a return connecting rod to a crosshead fitted to the two pump rods.

The condensers are cylindrical and built of thin sheet brass. They are placed in the wings of the engine rooms as low down as practicable. The circulating water is forced through them by means of centrifugal pumps, assisted by external scoops, which are so efficient as to render the services of the pumps almost unnecessary when the vessels are steaming at a high speed. The crankshafts, as well as the straight shafting, are of hollow steel. Balance weights have been fitted to the crank arms with a view to reduce the vibration of the engines.

The boilers are eight in number, and are of the Yarrow type. Four are placed in each of two boiler rooms, which are separated by a water tight bulkhead. The feed water is supplied to each boiler by an independent direct acting duplex Worthington pump. In addition to the four main pumps in each boiler room, an auxiliary pump is fitted which is capable of feeding any one or all of the boilers through the auxiliary service. These pumps are also arranged to deliver through the fire service on deck. This arrangement for feeding the boilers is reported to have worked in a very satisfactory manner, no difficulty having been experienced throughout in maintaining a steady level of water in the boiler, in spite of the fact that no automatic feed regulating gear was fitted.

The forced draft is supplied by two Belliss' fans, driven by independent engines fitted in each stokehold. The air from them reaches the fires by way of orifices in a screen built across the boiler room, and guarded by automatic flaps. The latter are designed to obviate danger in the stokehold through flames

caused by back draft, in turn the result of the bursting of a tube or some other cause. The boilers were designed for a working pressure of 185 pounds per square inch, and this pressure was maintained with a moderate air pressure in the stokehold throughout the official high speed trials.

The results of the official trial of the *Snapper*, being the means for three hours, were as follows. Steam pressure in boilers, 180 pounds per square inch; vacuum, 25 inches; I.H.P., 3,699; speed, 27.56 knots; air pressure in fire rooms, 2.66 inches. The mean speed for the six runs on the measured mile was 27.908 knots.

A satisfactory feature of these trials was the low power required to give the speed (*Salmon*, 3,589 to 3,628 I.H.P., and 27.608 knots; *Snapper*, 3,699 I.H.P., 27.56 knots). Careful observations which were made are reported to have shown that the wave formation was remarkably slight, and the alteration of trim between the vessels being at rest and at the high speed was unimportant, the squatting frequently observed in vessels of this class being scarcely noticeable. The propellers used were three-bladed, the blades being separate Stone's bronze castings secured to cast steel bosses by means of a special arrangement of wedges and bolts. It may be remarked that the propellers used on the successful trials were those originally designed, no alteration having been found necessary or desirable.

The following are the reported results of the twelve hours' consumption trial of the *Snapper*. Vacuum, 25.14 inches; I.H.P., 491; speed, 12.984 knots; air pressure, .35 inch; coal per I.H.P. per hour, 1.8 pounds.

This trial took place in the passage from the builders' yard on the Humber to Chatham, at which dockyard the vessels were delivered, four boilers only being under steam throughout the run.

The manœuvring trials and the usual stopping and starting trials were successfully made, the time occupied when the engines were going full speed ahead to bring them to full speed astern being six seconds. This is a satisfactory performance, especially in view of the fact that the vacuum in the condensers was de-

stroyed through a misunderstanding, one of the cocks on the condensers being opened at the same time that the order to go astern was received in the engine room.

FRANCE.

Charles Martel.—This battleship, which was launched at Brest in September, 1893, has made several preliminary trial trips.

One trial, at 71 revolutions per minute, gave a speed of 13.74 knots, with 5,474 I.H.P., the coal per I.H.P. per hour being 1.76 pounds.

Another, at 89 revolutions, gave a speed of 16.8 knots, with 10,839 I.H.P., the coal per I.H.P. being 2.3 pounds.

On August 11 a trial was made at 52.5 revolutions, the speed being 10.6 knots, I.H.P., 2,120, coal per I.H.P., 1.72 pounds, and coal consumed per square foot of grate surface, 10.65 pounds.

On August 15, a trial at about 4,900 I.H.P. was commenced, but was interrupted by the discovery that three steam pipes of the starboard engine were cracked. After these repairs had been completed, a six hours' coal consumption trial was made on September 30, the I.H.P. being 5,308, the coal per I.H.P. per hour, 1.41 pounds, and the speed, 13.8 knots.

St. Louis.—This first class triple screw battleship, a sister vessel of the *Charlemagne*, described on page 427 of Volume VI and page 823 of Volume VII, was successfully launched on September 8 at the Caudan Works, Lorient. The rate of building this battleship is remarkable for the French Navy, as the keel was laid only fifteen months before. The principal dimensions of this vessel are here repeated.

Length, 385.5 feet; beam at water line, 66.47 feet; draught, aft, 27.56, mean, 25.92 feet; displacement, 10,951 tons; I.H.P. at 125 revolutions per minute, 14,300, under forced draft; corresponding speed, 18 knots; speed with 9,370 I.H.P., 16.5 knots; normal coal supply, 666 tons. The engines are of the vertical four-cylinder triple expansion type. The twenty Belleville boilers, arranged in four compartments, have a working pressure of 240 pounds per square inch. There are two smoke pipes.

The *Charlemagne* and *St. Louis* are the first French battleships to use harveyized steel armor. This is used for the central part of the vessel. At the ends, the armor will be of special nickel or chrome steel. The details of the construction and the armament have already been given.

Gaulois.—This first class triple screw battleship, a sister vessel of the *St. Louis*, was successfully launched at Brest on October 6. As the construction of this vessel was commenced on January 22, 1896, the rate of building has exceeded that for the *St. Louis*. About 3,640 tons of material had been worked into the hull at the date of launching, an amount which is greater than the launching weight of the *St. Louis*.

Amiral Pothuau.—This armored cruiser, the building of which was begun at the end of 1892 by the Forges et Chantiers de la Mediteranée, near Havre, was launched, after a first unsuccessful attempt, on August 22d, 1895. The vessel arrived at Cherbourg on August 12th, 1896, to receive her armament and prepare for the official trials.

The designs of this cruiser are by Naval Constructor Thibaudier. Her length is 370.7 feet; beam, 50.2 feet; draft, aft, 21.32 feet; displacement, 5,236 tons.

The armor belt is of hardened steel, varying in thickness from 1.26 to 2.36 inches. The protective deck is 3.35 inches thick on the sloping sides and 1.38 inches thick on the horizontal part. The conning tower is .94 inch thick. There are two closed turrets, one forward and the other aft, placed on the center line of the vessel, and protected by 7.1-inch armor.

The armament consists of two 7.64-inch, 40-caliber guns, one in each turret, the turrets being worked either by electricity or by hand. Ten 5.45-inch rapid fire guns, eight of these in the armored redoubt, arranged with direct ahead or astern fire for four guns; the remaining two in sponsons, with an angle of fire from right ahead to astern. All of the 5.45-inch guns are protected by shields of special steel, 2.13 inches thick, this being the only protection for the two guns in the sponsons. There are, besides, sixteen 1.85-inch and eight 1.46-inch rapid fire guns, and five torpedo tubes.

The machinery consists of two vertical, inverted, triple expansion engines, designed for about 7,400 I.H.P. with natural draft, and about 9,860 with forced draft. The speeds expected are 18 and 19 knots, respectively. The boilers are of the Belleville type. The bunker capacity is about 530 tons.

Descartes.—Further trials of this cruiser were made on July 31 and August 11. The results of previous trials were given on page 626 of the August number.

The trial on July 31 was made under natural draft, the revolutions per minute being 123; speed, 18.2 knots; I.H.P., 6,194; coal per I.H.P. per hour, 1.77 pounds; coal burned per square foot of grate surface, 17.41 pounds. The engines worked well.

On August 11 the coal consumption trial was made, the speed obtained being 10 knots; revolutions per minute, 62; I.H.P., 902.5; coal burned per square foot of grate surface per hour, 8.6 pounds; coal per I.H.P. per hour, 1.71 pounds.

Extensive changes in the hull of this vessel are to be made at Brest, with a view to increasing her stability. This work will take from five to six months. According to "The Engineer," London, 125 tons of sand, of which 90 tons will be cemented, are to be placed in the double bottom.

Pascal.—This sheathed, twin screw, second class cruiser, described on page 189 of the February number, made a three hours' coal consumption trial on September 30. The I.H.P. was 7,133; coal per I.H.P. per hour, 1.87 pounds; coal burned per square foot of grate surface, 19.05 pounds; mean speed on measured mile, 18.5.

Catinat.—This sheathed second class cruiser, described on page 205 of Vol. VI, was successfully launched by the Société des Forges et Chantiers de la Mediterranée, Havre, on October 8. This cruiser is a modified *Descartes*, and is expected to make 19 knots. The principal data of the vessel as finished are as follows, according to "Le Yacht," Paris.

Length, 332 feet; beam, 44.62 feet; draught, aft, 21.16 feet; displacement, 4,001 tons.

The armament will consist of four 6.46-inch guns protected by

2.13-inch shields; ten 3.28-inch guns, similarly protected; ten 1.85-inch and four 1.46-inch guns, and two torpedo tubes.

The twin screw engines are designed for about 6,900 I.H.P. with natural, and 8,877 with forced draft. The speed at full power is expected to be 19 knots. The boilers are of the Belleville type.

The cost of the vessel complete will be about $1,615,860, of which about $249,327 will be for the guns and torpedoes. The vessel is to be delivered in February, 1897.

Durandal and **Hallebarde.**—These two torpedo boats are to be built by Mr. Normand, at Havre. The displacement will be about 290 tons, and the speed, 26 knots. The boilers will, of course, be of the Normand water tube type.

Cyclone.—This torpedo boat will also be built by Mr. Normand. The displacement will be about 145.26 tons, and the speed, 30 knots.

GERMANY.

Torpedo Boats.—Six sea-going torpedo boats are building by Mr. F. Schichau, Elbing. They will be 152.5 feet long between perpendiculars. The engines will be of a new type, having a simplified Joy valve gear, and are designed for 2,466 I.H.P.

There will be two Thornycroft water tube boilers with a total heating surface of 602.8 square feet. The guaranteed speed is 25 knots.

ITALY.

Carlo Alberto.—This armored cruiser, of the *Garibaldi* type, and a sister vessel to the *Vettor Pisani*, described on page 830, Vol. VII, was successfully launched on September 23. The launch of this vessel was previously reported as having taken place in March; from another report it seems that the vessel was launched in September, after a serious hitch. There is no reliable report of what happened during the intervening six months. The following data are from the "Rivista Marittima," Rome.

The length between perpendiculars is 324.8 feet; beam, 59.16 feet; draught, mean, 22.96 feet; displacement, 6,397 tons; I.H.P., 12,822; speed, 20 knots. The hull was built at the dockyard at Spezia, and the machinery by Messrs. Ansaldo & Co., Sampierdarena.

The armament will consist of twelve 6-inch Armstrong guns; six 4.7-inch and ten 2.24-inch guns; a number of smaller guns, and five under water torpedo tubes. The complement will consist of from 20 to 25 officers and 460 men. The protection and internal arrangements of the hull, the machinery and the coal supply are identical with those of the *Garibaldi*, described on page 605 of the August number.

The keel of the *Carlo Alberto* was laid on February 1, 1893.

NORWAY.

Valkyrien.—This torpedo boat destroyer, built by Mr. F. Schichau, Elbing, Germany, completed her trials on May 11.

The length is 190.2 feet; beam, 24.27 feet; depth, 13.45 feet; draught, 9.19 feet; displacement, 374 tons.

The armament consists of two 2.76-inch guns, two machine guns and two tubes for 17.72-inch Schwartzkopff torpedoes, one forward and the other aft.

There are twin screw engines, designed for 3,250 I.H.P., and two Thornycroft boilers, with 6,997 square feet of heating surface and a working pressure of 200 pounds per square inch. The propellers are 7.22 feet in diameter, the pitch being variable from 10.5 to 11.5 feet. The coal capacity is 88.5 tons.

Without forcing, an average speed of 23.2 knots was maintained for three hours, or 1.2 knots over the contract speed. The revolutions per minute were 240 and the I.H.P., 3,255.

This destroyer was paid for by the voluntary contributions of the Norwegian Women's Patriotic Association (Mittheilungen aus dem Gebiete des Seewesens, Pola).

PORTUGAL.

New Cruisers.—According to "Industries and Iron," London, the bids for one 3,550-ton and two 1,770-ton cruisers were as follows.

3,550-ton Cruiser.—Lowest bid by Mr. F. Schichau, Danzig, Germany, $1,097,500, without armament. Germania Co., Kiel, Germany, $1,107,500, without armament. The Naval Construction and Armaments Co., Barrow-in-Furness, $1,185,000, without

armament. Messrs. Armstrong, Mitchell & Co., Newcastle-on-Tyne, $1,418,000, including armament.

The general descriptions of this cruiser, taken from the proposals by the "Rivista Marittima," Rome, are as follows.

The hull of the vessel is to be of steel, wood sheathed and coppered, to be well subdivided, and to have a double bottom under the machinery spaces, at least. The load displacement to be about 3,550 tons on a maximum draught of 19.68 feet.

The armament to consist of rapid fire guns only, four 5.9-inch, eight 4.7-inch, twelve 1.85-inch, and six 1.46-inch. There are to be, in addition, four machine guns and five torpedo tubes, three of these to be under water.

The protective deck to extend the whole length of the vessel and to be from 4.33 to 1.57 inches thick. Cofferdams to be built above the protective deck on each side of the machinery spaces. There are to be two military masts. Provision to be made for a complement of about 300. To have twin screw vertical, triple expansion engines and water tube boilers. The speeds with natural and forced draft to be 20 and 22 knots, respectively, maintained continuously for six hours. The normal coal supply to be about 690 tons, with provision for not less than about 290 tons additional.

Two 1770-ton Cruisers.—Lowest bid by the Thames Shipbuilding & Iron Co., $440,000 each. Germania Co., $487,500 each. Mr. Schichau, $640,000 each.

These cruisers are also to be of steel, sheathed, with a displacement of about 1,770 tons on a maximum draught of 13.78 feet.

The armament to consist of two 5.9-inch, four 4.7-inch and eight 1.85-inch rapid fire guns, two machine guns and one torpedo tube.

To have vertical triple expansion engines and water tube boilers, the speed for six hours to be 15 knots under natural draft. The coal supply to be the greatest possible. The complement will be about 200.

It is not yet known whether these vessels are to be built.

Adamastor.—This cruiser, described on page 606 of Volume VII, was successfully launched by Messrs. Orlando Bros., Leg-

horn, Italy, in July. The keel was laid in June, 1895. The length over all is 259.5 feet.

RUSSIA.

Rostilav.—This steel battleship, building for the Black Sea fleet, was successfully launched at Nicolaieff on September 2. Her length is 345 feet; beam, 67.6 feet; draught, aft, 23 feet, mean, 22 feet; displacement, 8,740 tons.

For about four-fifths of her length there is a belt armor, about 7 feet wide and about 16 inches thick in the middle, and 11.8 inches at the ends. Above this belt there is a central citadel 5 inches thick. The protective deck, varying in thickness from 2 to 2.95 inches, meets the belt armor about 5 feet below, and rises in the center to about 2 feet above the water line.

The armament consists of four 12-inch 35-caliber guns, mounted in pairs in turrets, the armor of which is from 9.8 to 11.8 inches thick. These guns have an angle of fire of 270 degrees. Six 6-inch rapid fire guns in the citadel; sixteen 1.85-inch and 1.46-inch rapid fire guns on deck and in the tops; and six torpedo tubes, one forward, one aft, and two on each broadside.

The twin screw, triple expansion engines are to develop about 8,400 I.H.P. and give the ship a speed of 16 knots. There are sixteen cylindrical boilers in four compartments. The furnaces are fitted for burning coal and liquid fuel.

SPAIN.

New Battleship and Cruisers.—Invitations to bid on one battleship and one cruiser were sent by the Spanish government to several leading shipbuilding firms in September, to which seven British firms responded. According to "Industries and Iron," London, the commission to examine these bids has reported in favor of Messrs. Armstrong, Mitchell & Co., Newcastle-on-Tyne, on the following general plans.

The battleship to be of about 11,000 tons displacement, with a speed of 19 knots. The main armament to consist of four 9.45-inch and sixteen 5.9-inch guns. The armor to be of harvey-

ized steel. The vessel to be completed in fourteen months at a cost of about $4,500,000.

The cruiser to be of 6,500 tons displacement and to have a speed of 21 knots. The main armament to consist of two 7.87-inch and eight 5.9-inch guns. The vessel to be completed in twelve months at a cost of about $3,000,000.

Water tube boilers will be used in both vessels.

Another cruiser of about 1,800 tons displacement and a speed of 20 knots is to be ordered in Great Britain.

Cristobal Colon.—This armored cruiser, which was building by Messrs. Ansaldo & Co., at Sestri Ponente, Italy, for the Italian Government, and bought by the Spanish Government, was successfully launched on September 16.

The hull and the twin screw engines are practically duplicates of the *Garibaldi*, described on page 605 of the August number.

The boiler installation was originally to consist of 16 Belleville boilers, but, according to the "Revista General de Marina," Madrid, and "Marine Française," Paris, 24 Niclausse boilers, which are to be made in Paris, are now to be put in. The following data of these boilers are from the "Marine Française."

Total grate surface, 948.24, total heating surface, 30,965 square feet; ratio of H.S. to G.S., 32 to 1. They will be arranged in equal groups in four compartments. Each boiler will consist of 11 collectors containing each 18 tubes, or a total for all boilers of 264 collectors and 4,752 tubes. The inside and outside diameters of the tubes are 2.95 and 3.23 inches, respectively.

The working pressure will be about 215 pounds per square inch, and the boilers are to be delivered in four months from the date of the order.

The weight of the 24 boilers, without water, will be 315.9 tons; with water, 375 tons; total with water and all accessories, 384.35 tons. Using the last weight, gives 27.8 pounds of boiler per square foot of H.S., 907.9 pounds per square foot of G.S., and 62.35 pounds per I.H.P. (13,808 full power). The I.H.P., full power, per ton of boiler will be 3.59.

The total floor space occupied by the boilers will be 1,358.42 square feet. The width of each boiler compartment, containing

six boilers, will be 16.08, and the length, 43.31 feet; of this space, the six boilers will occupy 339.6 square feet. The I.H.P. at full power will be .838 per cubic foot of space occupied by boilers. There will be two smoke pipes with a total area of 141.2 square feet, the height above grates being 78.74 feet. The closed fire room system of forced draft will be used, eight blowers being capable of maintaining an air pressure of about one inch.

The normal coal supply will be about 590 tons; this can be increased to 1,180 tons.

The complement will consist of 500 officers and men.

Princesa de Asturias.—This armored cruiser, building at Carracas, near Cadiz, Spain, was to be launched on October 9, but she stuck in the mud. On October 17 she slipped into the water without help, and, apparently, uninjured. The vessel was ordered in September, 1889.

The length is 347 feet 10 inches; beam, 61 feet; depth, 21 feet 10 inches; displacement, about 6,890 tons. The hull is of steel and protected by a water line belt and a protective deck. There will be two smoke pipes and two light masts.

The armament will consist of two 9.45-inch guns, in barbettes, one forward and the other aft, and ten 5.51-inch guns; these twelve are Hontoria guns. Eight 2.24-inch and four 1.65-inch Nordenfeldt guns; eight 1.46-inch Maxim guns and eight torpedo tubes.

There will be twin screw, triple expansion engines, which are to develop about 14,790 I.H.P. and to give the vessel a speed of 20 knots. The coal supply will be about 1,180 tons, and the complement 500 men.

Dona Maria de Molina.—This torpedo gunboat was successfully launched at Ferrol on October 9.

The length is 232.9 feet; beam, 27.23 feet; draught, aft, 8.76 feet; displacement, 817 tons.

The armament will consist of two 4.7-inch Hontoria guns, one forward and the other aft; four 1.65-inch Nordenfeldt rapid fire and two Gatling guns; and four under-water torpedo tubes.

The twin screw, triple expansion engines are designed for 4,537 I.H.P. and to give the vessel a speed of 20 knots. The coal sup-

ply is about 120 tons. The complement will be 89 officers and men. The keel was laid on September 29, 1894.

Furor and **Terror.**—These torpedo boat destroyers, built by Messrs. J. and G. Thomson, Clydebank, were launched in August and September, respectively, with all machinery in place. The length is 220 feet; displacement, ready for sea, 380 tons; speed, 28 knots.

TURKEY.

Pelenkdideria (**Sea Tiger**).—This torpedo cruiser, built by the Germania Co., Kiel, Germany, was accepted after completing her trials.

The length is 236.2 feet, and displacement, 827 tons. The twin screw, triple expansion engines were designed to develop 4,932 I.H.P. and give the vessel a speed of 20 knots.

MERCHANT STEAMERS.

Creole.—This passenger and freight steamer, built by the Newport News Shipbuilding and Dry Dock Co., for the Cromwell Line, was successfully launched on August 8. The designs of the vessel are by Mr. Horace See, consulting engineer of the steamship company. She will run between New York and New Orleans. The following description is compiled from the "Marine Journal" and "Seaboard," New York.

Her principal dimensions are as follows: Length over all, 375 feet; breadth, 44 feet; depth, 32.5 feet.

The hull is built entirely of steel, with three continuous decks and a partial orlop deck forward and a promenade deck at top of main deckhouse extending about 170 feet in length. The *Creole* is the first vessel sailing out of New York in the coastwise trade to be fitted with a water bottom. Besides this, her hull is subdivided into a number of water tight compartments by means of transverse bulkheads. The outside plating has vertical lap joints below the water line. She is rigged with two steel pole masts and the necessary booms for handling cargo.

The *Creole* has accommodation for 75 first class passengers amidships, in the most comfortable part of the ship, and for 150 steerage passengers aft. The large dining saloon is forward and extends the full width of the deck house. It has extra large round sidelights, giving an unobstructed view forward and on both sides of the ship. The general arrangement of the passenger accommodation is, in many respects, a new departure from the usual type of our coastwise passenger steamers, and is more on the plan of the modern transatlantic steamship. The social hall is fitted with piano, library, &c. Both dining saloon and social hall have electric ventilating fans. The smoking room for first class passengers at after end of promenade deck is large and fitted with all the conveniences of an ocean-going vessel.

There are two large lobbies, one on upper deck and one on promenade deck, connected by the main companionway. All state rooms for first class passengers and officers have good light and are well ventilated. The vessel is lighted throughout by electricity, and there is a powerful searchlight on the foremast. All running lights are connected to a Russell See Electrical Indicator in the pilot house. Life boats, rafts, &c., of the latest design are fitted according to the United States Steamboat Inspection Laws.

There is a steam steering gear, made by Messrs. Williamson Bros., Philadelphia; hoisting engines, Providence steam windlass and capstans, and a complete outfit of pumps, made by the George F. Blake Manufacturing Company, New York, of their latest vertical duplex (Admiralty) type, such as is used in the United States cruisers.

The main engine is of the vertical triple expansion type, with cylinders 28, 44 and 74 inches in diameter and a stroke of 54 inches. The valves, of the piston type, are worked by the See-Marshall valve gear, each valve receiving its motion from a separate eccentric. There are three cylindrical, double-ended steel boilers, each having six corrugated furnaces with a common combustion chamber. The working pressure is 180 pounds. The centrifugal circulating pump is driven by an independent engine. The See Hydro-Pneumatic Ash Ejector will be fitted in each fire room for discharging ashes.

In the construction of the vessel particular attention has been paid to the comfort of the passengers and every effort made to have her first class in every respect, while her staunchness is enhanced by being designed under the rules of the American Shipmasters' Association, and the close inspection of its officers.

L. C. Waldo.—This lake steamer, built by Messrs. F. W. Wheeler & Co., West Bay City, Mich., and fully described on page 423 of the May number, has been made the subject of tests by Mr. G. C. Shepard, Mechanical Engineer. The following data are taken from the report of these tests, published in the "Marine Review," Cleveland.

Five tests were made, varying in duration from 3 hours and 47 minutes to 5 hours and 57 minutes, four of the tests being for 5 hours or over. One of the tests being incomplete, it has been omitted from the table. The feed water was measured by a Union meter, which had been calibrated and found to be correct to within 1 per cent.

All the coal used during tests was carefully weighed, and the time of starting and stopping each test was so arranged that the fires were all cleaned once during the test, and were in the same condition at the end as at the beginning. In the shortest test, No. 3, these precautions were observed, but on account of the short duration of the test it did not permit the burning down of the extra amount of coal used in cleaning the fires, so that the fires were not in the normal condition at the end of the test. The ashes, weighed on the longest test, No. 2, were found to be 12.7 per cent. of the coal, and this amount has been used for all the tests in determining the amount of combustible. Observations were made and indicator cards taken every hour.

Number of test.	1	2	3	4
Displacement, net tons	4,258	6,513	6,453	6,446
Area, immersed midship section	459.2	685.2	681.2	681
Mean boiler pressure, pounds	166	165.5	166.8	166.4
I.P. receiver pressure, pounds	55	56.3	55.7	62
L.P. receiver pressure, pounds	8.03	8.2	9.06	9.4
vacuum, inches	25.5	25.4	25.8	25.3
revolutions, main engine	80.77	83.48	82.34	87.63
fan engines	265	270	267	361
Indicated horse power	1,547.7	1,748.4	1,704.2	2,066.4
Speed, miles per hour	14.21	13.3	13.04	14.02
Slip, per cent	11.5	19.8	20.3	19.5
Temperature of feed water	176	183.5	175	174
at furnace	235	285	365	390
fan	.....	405	530	590
Draft at furnace, inch	.21	.23	.5	.25
fan, inches	.....	1.55	1.65	2.5
Coal per hour, pounds	2,715	3,376	3,711	3,546
I.H.P.	1.75	1.92	2.17	1.71
Water per I.H.P. per hour, pounds	16.13	17.03	17.69	16.45
pound of coal per hour, pounds	9.2	8.82	8.18	9.52
combustible, pounds	10.53	10.10	9.29	10.98
from and at 212°, per pound of combustible, pounds	11.45	10.91	10.12	11.97

The exhaust from all the auxiliaries passed through the tubes of a feed water heater, and raised the temperature of the feed water about 60 degrees. The average temperature of the injection water was about 54.5, and of the hot well, 119 degrees.

There are two boilers with a total grate surface of 120, and a heating surface of 6,230 square feet, and fitted with Ellis & Eaves' suction draft and plain tubes.

No air was admitted above the grates, the valves for this purpose being shut during all the tests. The dampers in the ash pit were closed except during test No. 3.

Kaiser Wilhelm der Grosse.—This is the name of the express steamer, building by the Vulcan Works for the North German Lloyd S. S. Co., and of which the general dimensions were given on page 648 of the August number. The following additional data have been kindly supplied by the Geo. F. Blake Manufacturing Co., New York, to whom the JOURNAL is also indebted for the cut of the air pumps built for this steamer.

The twin screw engines are of the four cylinder, triple expansion type, with cylinders 52, 90 and 97 (2) inches in diameter, the stroke being 69 inches. The working pressure in boilers will be about 185 pounds per square inch. The engines are designed for 77 revolutions per minute, with cut off at .72 of the stroke.

The twin air pumps, one for each engine, are of the Blake type and are similar to those on the U. S. S. *Minneapolis* and *Brooklyn*, except that slides and guides have been added on account of the large size of these pumps. A detailed description and sectional cuts of the *Minneapolis*' pumps were given on page 708 of Vol. VI.

The pumps of the German steamer have each two double-acting steam cylinders, 18 inches in diameter, and two single-acting air cylinders, 44 inches in diameter, the stroke being 24 inches. At a test of these pumps, made at the Blake Company's Works at East Cambridge, Mass., they formed a vacuum of 28.75 inches, the temperature of the vapors and condensed water being about 80 degrees. The pump was run as high as 75 double strokes per minute, and ran as smoothly and quietly as at one double stroke per minute. Although the pump was only placed

BLAKE
BLAKE PATENT
VERTICAL TWIN AIR PUMP
18" 44" 24"
for the German Lloyd Express Steamer
of 28,000 I.H.P.

on and not bolted to the floor, it worked without any vibration, showing that it is self-contained and the parts well balanced.

The feed, fire, bilge and water service pumps of this steamer are all of the Blake vertical type.

Kaiser Friedrich.—This is the name of the express steamer building by Mr. Schichau at Danzig for the North German Lloyd Steamship Company, and which was described on page 646 of the August number.

Barbarossa, Bremen, Friedrich der Grosse and **Königin Luise.**—These twin screw freight steamers are building for the North German Lloyd Steamship Company.

They are to be 525 feet long, with a beam of 60 feet and a depth of hold of about 36 feet. The displacement given ranges from 17,600 to 20,000 tons. The dead weight capacity is about 10,500 tons. The *Barbarossa*, building by Messrs. Blohm & Voss, Hamburg, was successfully launched on September 5. The *Bremen* is building by Mr. F. Schichau, Elbing, and the *Friedrich der Grosse* and *Königin Luise*, by the Vulcan Works, Stettin. The last named vessel was successfully launched on October 10.

The twin screw engines are all to be of the vertical, inverted, quadruple expansion type, those for the *Barbarossa* of 8,000 I.H.P., and the others of 7,000 I.H.P., the expected speeds being 15 and 14 knots, respectively.

The arrangement of cylinders on the *Friedrich der Grosse* and *Königin Luise* is as follows. First comes the H.P. piston valve, its center line being in the vertical plane through the axis of the shafting, and then the H.P. cylinder; next comes the L.P. cylinder, with a slide valve on its front; then the second I.P. cylinder, also with a slide valve on its front; then the first I.P. cylinder, with its two piston valves between it and the second I.P. cylinder. The slide valves will be worked by rocking levers from the eccentrics. The engine frames, front and back, will be of the usual cast pattern. The connecting rods will be very light. The engines are to run at 75 revolutions per minute.

There will be five double-ended and two single-ended boilers, the working pressure being 220 pounds per square inch. The *Bremen* will be fitted with Howden's forced draft system.

Besides the above, this company has ordered three smaller steamers, which are to be 325 feet long and have a tonnage of about 4,500. One is building by Messrs. Blohm and Voss, Hamburg, another by Mr. J. Tecklenborg, Geestemünde, and the remaining one by the Seebeck Co.

Pennsylvania.—The first of two large twin screw steamers building for the Hamburg American Steam Navigation Co., was successfully launched by Messrs. Harlan and Wolff, Belfast, Ireland, on September 10. Some particulars of the *Pennsylvania* were given on page 851 of Vol. VII.

The launching weight was about 8,000 tons. The I.H.P. of the two engines will be from 6,000–7,000, and the speed of the vessel, from 14 to 15 knots. The shafting is said to be 230 feet long.

Another steamer of the same dimensions, drawing about 29 feet when fully loaded, has been ordered of Messrs. Blohm and Voss, Hamburg, Germany. The engines will be of Schlick's patent quadruple expansion type, and are designed for about 5,500 I.H.P.

Augusta Victoria.—This fast passenger steamer, of the Hamburg American Steam Navigation Company, which has been running on the transatlantic route for about seven years, will, according to "Seaboard," New York, be lengthened 60 feet, have an additional boiler put in and one of her three masts removed. The alterations are to be made this winter by Messrs. Harlan & Wolff, Belfast, Ireland. It is expected that the increased length will give accommodation for 100 additional first cabin and 200 additional steerage passengers, and that the cargo capacity will be enlarged by about 500 tons. The extra length to be added will make the vessel 520 feet long on the water line. The vessel will be cut in two between the second and third boiler compartments, and the forward compartment will be inserted at that point. An extra boiler will be added, making nine main boilers in all. With this additional boiler power, a corresponding gain in speed is expected, as the engines are large enough to utilize the extra supply of steam. The space now occupied by the forward boiler compartment will be used partly for state rooms.

As no description of the *Augusta Victoria* has heretofore been published in the JOURNAL, the following data may be of interest.

The vessel was built by the Vulcan Works, Stettin, Germany, in 1889. The length is 480 feet over all and 460 feet between perpendiculars; beam, 56 feet; depth of hold, 39 feet; mean draught, full load, 24 feet; displacement, 10,000 tons. The hull is of steel.

The twin screw engines are of the inverted, three cylinder, triple expansion type with cylinders 41.3, 66.9 and 106.3 inches in diameter, the stroke being 63 inches. All cylinders are fitted with piston valves having balance pistons. Each H.P. cylinder has one, and each I.P. and L.P. cylinder has two valves, the diameter of the H.P. and I.P. valves being 23.62 inches, and of the L.P. valves, 37.4 inches. The valve gear is of the Stephenson link type, the eccentrics being secured to the crank shaft couplings. The steel crank shafts are 19.69 inches in diameter and have a 4.72-inch axial hole; the crank pins are of the same diameter, the hole being 3 inches in diameter.

The condensers have a total cooling surface of 18,944 square feet, the tubes being $\frac{3}{4}$ inch in outside diameter. The air and circulating pumps are independent.

There are eight double-ended, steel main boilers, each 15 feet $1\frac{1}{4}$ inches in diameter and 19 feet $6\frac{3}{4}$ inches long, and one auxiliary boiler. There are six corrugated furnaces in each main boiler, the least internal diameter being 45.75 inches. There is one combustion chamber for each furnace, an iron plate, protected on both sides with fire brick, making the division between opposite furnaces. The front connections are also separate. The total grate surface of the main boilers is 1,120, and the heating surface, 36,000 square feet. The tubes have an external diameter of 3.33 inches. The boilers are arranged in three compartments, with athwartship fire rooms, three main boilers in each of the two after compartments, and the auxiliary and two main boilers in the forward compartment.

The propellers are of steel, with four blades. The diameter is 18, and the pitch 32 feet.

The maximum I.H.P. is about 25,000. The bunker capacity is 2,250, and the daily consumption of coal, about 220 tons.

Alexander Holly.—This steamer, built by the American Steel Barge Co., West Superior, Wis., for the Bessemer Steamship Co., was launched on August 12. Her length is 379 feet; beam, 46 feet; depth of hold, 26 feet; gross carrying capacity, 4,150 tons on a draught of 14.5 feet.

The machinery is practically the same as that of the *John Ericsson*, described on page 651 of the August number. The steam steering apparatus and the three steam capstans were furnished by the Bath Iron Works, Bath, Maine.

Sir William Fairbairn and **Robert Fulton.**—These steel steamers, built by the Detroit Dry Dock Co., Detroit, Mich., at their Wyandotte yard, for the Bessemer Steamship Co., were successfully launched on August 1 and September 10, respectively. The following particulars are from the "Marine Record" and "Marine Review," Cleveland, Ohio.

Length over all, 438 feet, between perpendiculars, 414 feet; beam, 45.5 feet; depth, 28 feet. The hull is built on the channel bar system, the water bottom being divided into eight compartments.

The single screw engine is of the triple expansion type, with cylinders 24, 38 and 64 inches in diameter, the stroke being 42 inches. There are two boilers, each 14.5 feet in diameter by 11.5 feet long, the working pressure being 165 pounds per square inch. Howden's forced draft system is fitted. The propeller is 14 feet in diameter.

The cost of each steamer is said to be $256,000.

George Stephenson.—This steel steamer, building by Messrs. F. W. Wheeler & Co., West Bay City, Mich., for the Bessemer Steamship Co., was successfully launched on September 23d. The following data and description are from notes kindly supplied by the builders and from the "Marine Record," Cleveland.

The length over all is 428 feet, on keel, 408 feet; beam, 48 feet; depth, 28 feet. The water bottom is 66 inches deep and divided into six compartments. The gross carrying capacity, on a draught of 14.5 feet, is estimated at 4,100 tons. The machinery

is well aft, the fire room being athwartship and forward of the boilers. The bunkers are forward of the fire room. The general arrangements and the requirements for materials are the same as for the other steamers of this company.

The engine is of the vertical, triple expansion type with cylinders 25, 40 and 67 inches in diameter, the stroke being 44 inches. The H.P. and L.P. cylinders have piston valves and the I.P., a double ported slide valve, all worked by the double bar Stephenson link motion. There are three 3-furnace boilers, each 13.5 feet in diameter and 12.5 feet long, fitted for natural draft, the working pressure being 175 pounds per square inch. The total heating surface is about 2,400 square feet. The furnaces are fitted with the Etna shaking grate bars. The propeller is 13.5 feet in diameter and has a pitch of 17.5 feet.

James Watt.—This steel steamer, building by the Cleveland Ship Building Co., Cleveland, Ohio, for the Bessemer Steamship Co., was successfully launched on September 23. The following particulars are from the "Marine Record," Cleveland, and "Seaboard," New York.

Length over all, 425 feet, between perpendiculars, 406 feet; beam, 48 feet; depth, 28 feet; gross carrying capacity on a draught of 14 feet, 4,000 tons, and 6,000 tons on 18 feet. The water bottom is 66 inches deep. The hull is built on the channel bar system.

The engine is of the vertical, triple expansion type with cylinders 25, 40 and 68 inches in diameter, the stroke being 42 inches. There are three boilers, each 13 feet in diameter and 13 feet long, the working pressure being 175 pounds per square inch. The steam steering gear was made by the Messrs. Williamson Bros., Philadelphia.

Corcovado.—This steel steamer, built for the Pacific Steam Navigation Company by Messrs. C. S. Swan & Hunter, Wallsend, England, was successfully launched on July 25, and completed her trials on September 16. The *Corcovado* is the largest cargo steamer in the company's fleet. The length over all is 405 feet; beam, 47.25 feet; depth, 32.75 feet; dead weight carrying capacity, about 6,600 tons. There are seven main water tight

bulkheads. The double bottom, which extends the whole length of the vessel, has ten longitudinal divisions.

The engine is of the triple expansion type, with cylinders 24, 40 and 66 inches in diameter, and a stroke of 51 inches. There are three cylindrical boilers, the working pressure being 190 pounds per square inch. The shafting is of steel and the propeller blades of bronze. The machinery was built by the Wallsend Slipway and Engineering Company.

The trials consisted of a series of runs over the measured mile, which gave a mean speed of 12.98 knots, and a run of 100 nautical miles, the vessel carrying a dead weight load of 4,600 tons. The mean speed obtained for the 100-mile run was 12.58 knots.

A duplicate of this steamer has been ordered by the same company from the builders of the *Corcovado*.

Dunvegan Castle.—The following details of the machinery of this Castle line steamer, described on page 655 of the August number, are from "Engineering," London:

The engines are of the inverted cylinder, three crank type, but the sequence of cylinders differs from the usual practice. The H.P. cylinder, placed forward, is 38 inches in diameter; the L.P. cylinder is in the center and 102 inches in diameter, and the I.P. cylinder, 62½ inches in diameter, is placed aft; the stroke being 66 inches. The cylinders are steam jacketed, their draining being effected by Laing's automatic trap. The pistons are of cast steel and conical; the piston rods, connecting rods and crossheads are of mild steel.

The reversing is effected by a Brown's combined steam and hydraulic engine. A similar and smaller engine is used for working the main equilibrium valve on the main engines. To the condenser, which is of cast iron, and placed at the back of the main engines, forming part of the framing of the main engines, are attached two air pumps, driven by levers, one from the crosshead of the H.P. and the other from the I.P. engine. From the air pump crosshead are driven the bilge, sanitary and hot water pumps.

The crank shaft is made in three parts and interchangeable. Each part consists of the coupling, two discs, the crank cheeks

and pin, all one forging. The crank shaft is 20½, the pins, 21, the tunnel shafting, 19¾, and the propeller shaft, 21 inches in diameter. The propeller is 19.5 feet in diameter, with movable bronze blades; the boss is of cast steel. The main feed pumps are Weir's type and separate, and draw from a large suction filter. There are two main centrifugal circulating pumps, each of sufficient size to do the whole work, and driven by a single double-acting engine. Placed at the back of the main engines, are one large ballast engine, and three donkey pumps for sanitary and general work. There is also one large special plunger pump for circulating the water in the boilers at full working pressure. There is an auxiliary condenser with a centrifugal circulating pump and a small duplex air pump. There are two large Weir's evaporators and donkeys for the necessary make up. At the side of the starting platform, and on the same level, are the electric light machines and the refrigerator.

Each of the three double-ended boilers is 15 feet 3 inches in diameter, and 19 feet 2 inches long, and has six Purves' furnaces 3 feet 9 inches in internal diameter. The two single-ended boilers are 16 feet 10 inches in diameter and 10 feet long, each having four Purves' furnaces 3 feet 7 inches in internal diameter. One single-ended boiler is placed in the forward and one in the after stokehold. There is one funnel, 12 feet 3 inches in diameter, for the five boilers, with an air downtake on the forward and the after sides. In each of these downtakes is placed a large fan, so arranged that it can send a large supply of air to the stokeholds when the steamer is in the tropics. The main steam pipes are of wrought iron. See's ash ejectors are placed in the stokeholds.

The engines indicated 8,000 H.P. on trial, and the vessel's speed was 17½ knots. She will average about 16 knots on the regular Cape voyage.

Andalusia.—Another of the steel emigrant and freight steamers, built by the Palmer Shipbuilding and Iron Company, Jarrow-on-Tyne, for the Hamburg American Steam Navigation Company, was successfully launched on August 25. She is a sister

vessel of the *Adria* and *Armenia*, described on page 648 of the August number.

Adria.—The trial of this steamer, described on page 648 of the August number, was made on July 28. A mean speed of 12.5 knots was attained on the measured mile, the revolutions being 75 per minute.

Ramapo.—This steel steamer, built by the Union Dry Dock Co., Buffalo, N. Y., was launched August 1. The following is from "Seaboard," New York.

The length over all is 340 feet; length on keel, 319 feet; depth, 27 feet 3 inches; beam 44 feet; sheer forward, 4 feet; sheer aft, 2 feet 6 inches. It is the intention of lengthening her 60 feet, when dock facilities in Chicago are adequate.

The engines were built by the King Iron Works, Buffalo, and are of the vertical, inverted, triple expansion type with cylinders 23, 38½ and 64 inches in diameter and 42 inches stroke.

The four boilers, built by the Lake Erie Boiler Works, Buffalo, are 11.5 feet in diameter and 13 feet long, each with two 44-inch furnaces. The working pressure is 175 pounds per square inch.

Between decks there is a double cylinder, vertical hoisting engine with cylinders 10 by 12 inches, with line shafting operating double drums for eight hatches. The electric light plant is in the engine room.

It is expected that this steamer will carry 4,000 tons on a draught of 16 feet. At this draught, the displacement will be 5,480 tons. The expected speed is 14 miles per hour. Her cost will be $220,000.

Great Western Railway Co.'s new Steamers.—This company has ordered two 20-knot, twin screw steamers from the Naval Construction and Armaments Co., Barrow, for the Channel Islands service. They will have a length of 280 feet; beam, 34 feet 4 inches; depth, 16 feet; draught, about 11 feet 4 inches; displacement, 1,600 tons. The triple expansion engines will have cylinders 23, 36 and 56 inches in diameter, the stroke being 33 inches. There will be two double-ended boilers, each 15 feet 2 inches in diameter and 19.5 feet long, the working pressure being 175 pounds per square inch. The total grate surface will be 244,

and the heating surface, 10,000 square feet. The I.H.P. of the engines will be 6,000, and the speed in service to be 20 knots.

Ignazio Florio.—This twin screw mail steamer, built by the Messrs. Orlando Bros., Leghorn, Italy, and partially described on page 209 of the February number, completed satisfactory trials, under the supervision of naval officers, in July. The following data of the hull, the machinery and the trials are from the "Rivista Marittima," Rome.

The displacement on a mean draught of 15.19 feet is 2,106 tons. At this draught, the area of the immersed midship section is 435.45 square feet and the wetted surface, 12,217 square feet. Coefficient of fineness, block, .512, of midship section, .863. The total bunker capacity is about 335 tons. The vessel in time of war will carry six 2.24-inch and two 1.46-inch rapid fire guns.

The cylinders of the triple expansion engines, as previously mentioned, are 23.23, 37.4 and 59.1 inches in diameter, the stroke being 36.03. The volume of I.P. cylinder divided by that of H.P. cylinder is 2.592; L.P. ÷ I.P., 2.493, and L.P. ÷ H.P., 6.463. The diameter of piston rods is 5.51 inches, and the length of connecting rods, 74.8 inches. The crank pins are 10.83 inches in diameter and 11.81 inches long. The crank shafts are 10.83 inches in diameter.

The air pumps are worked from the main engines, and have a diameter of 21.26 inches and a stroke of 20.28 inches. The circulating pumps are independent and of the centrifugal type, the diameter of the vanes being 29.92 inches, and the discharge capacity of the pumps about 500 tons per hour. The feed pumps are independent.

There are two double-ended steel boilers in one compartment, with four Fox's corrugated furnaces in each end. Ferrando's grate bars are used, and two forced draft blowers are fitted. The working pressure in the boilers is 155 pounds per square inch. There is, besides, a small single-ended boiler for auxiliary purposes, situated abaft the main boilers. The main boilers are 14.75 feet in diameter and 22.75 feet long. The least internal diameter of the furnaces is 36 inches. The total grate surface is 318.6, and the heating surface, 10,711 square feet. Ratio of

H.S. to G.S., 33.62 to 1. Steam room in each boiler, 678 cubic feet.

The auxiliary boiler is 7.58 feet in diameter and 7.64 feet long. There is one smoke pipe.

The propellers are three bladed, the diameter being 11.15, and the mean pitch, 17.97 feet; ratio of pitch to diameter, 1.61 to 1.

Total weight of machinery, with accessories and water, 501.9 tons. Weight per I.H.P., developed on trial, 271.2 pounds.

The trial was made in calm weather and a smooth sea. The speed was obtained from two runs over a measured base of 31.39 nautical miles. The mean draught, measured just before leaving Leghorn, was 15.19 feet, the corresponding displacement being 2,106 tons. The mean results of the trial, which lasted four hours, were as follows.

Steam pressure in boilers, about 153 pounds per square inch; vacuum, 26.5 inches; revolutions per minute, 122.5; I.H.P., starboard, 2,041, port, 2,104, total, 4,145; speed, 17.95 knots.

Ernest Bazin.—This steamer, the first one built on the roller idea of Mr. Bazin, was launched at St. Denis, France, on August 19. The following particulars and description are from "Le Yacht," Paris.

The length of the vessel is 126.3 feet, and beam 37.73 feet. Flotation for the superstructure is secured by six upright, lenticular wheels or rollers, each 32.8 feet in diameter and 11.8 feet wide or thick at the center. The wheels are arranged in pairs, each pair connected by a steel shaft or axle. There are four journals on each shaft, making twelve supports in all for the superstructure, which contains the motive machinery and the living accommodations. This superstructure consists of four parallel girders, each 5.9 feet deep, strongly braced and tied to each other. A deck covers the top of this platform.

The motive machinery is divided into two distinct parts, one engine of about 500 H.P. working a three-bladed bronze propeller, and three engines, each of about 45 H.P., for turning the wheels. To obtain the maximum efficiency, it is necessary that these wheels should be turned at a speed proportional to the forward speed of the whole vessel.

The propeller is at the end of an oblique shaft. This arrangement is disadvantageous, but made necessary because the engine is so far above the water level. By placing the engine forward and as low as possible, the angle of obliquity of the propeller shaft has been reduced to 10 degrees, making a fair efficiency probable. Mr. Bazin, has, however, always believed that with large vessels of this type, paddle wheels should be used instead of the screw propeller. As the *Ernest Bazin* is only an experimental vessel, the small and possible reduction in speed due to the obliquity of the propeller shaft, will not interfere with the test of the correctness of the inventor's principle.

The fact that the habitable part of the vessel is distinct from what may be called the hull causes difficulties in arrangement, of which the one given above is the principal one. Another difficulty, the supply of sea water for the engines and other purposes, has been overcome by the constructor. Mr. Dubar, director of the Cail works, conceived the idea of utilizing the stern post, which is necessarily in the water, for this purpose. The stern post is a pointed tube, open at its base and solidly braced to the superstructure, through which the water rises part way by virtue of the speed of the vessel. A suction pump on deck raises the water the rest of the distance.

The displacement of the *Bazin* is about 242 tons. If the results come up to the expectations of the inventor, a speed of from 22 to 25 knots will be attained, with a total horse power of about 650. The combined immersed midship section of the six wheels is 419.8 square feet; to drive this same section at 25 knots in a vessel of the ordinary form would require an expenditure of at least 9,370 horse power. This method of comparison may seem exaggerated for the reason that the real cross section of the roller vessel is that of only one pair of wheels—that is, 139.9 square feet. But even with this reduced figure, the corresponding horse power for an ordinary type of vessel would be 3,126, or about five times that of the *Bazin*.

Should the system be a success, an enormous economy in coal consumption would be obtained. As the displacement is so

small compared to the weight of the total structure, this type of vessel could never be used as a cargo steamer.

John Englis.—This steel passenger and freight steamer, building by the Delaware River Iron Shipbuilding and Engine Company, Chester, Pa., was successfully laûnched on October 22. A few particulars of this vessel were given on page 430 of the May number, to which may be added the following data, which were kindly supplied by Mr. Randall T. Dickinson, the superintending constructor of the company.

The length over all is 313 feet 4 inches; beam, 46 feet; depth of hold, 17.5 feet.

The triple expansion engine has cylinders 30, 48 and 75 inches in diameter, the stroke being 54 inches. They are designed to develop 4,000 I.H.P. with 90 revolutions per minute, giving the vessel a speed of 16 knots.

There are six cylindrical, single-ended steel boilers, each 13 feet 2 inches in diameter and 12 feet 2 inches long, and fitted with three Purves' ribbed furnaces, 40 inches in internal diameter. The working pressure in boilers will be 180 pounds per square inch.

The propeller is of cast steel, four bladed, 15 feet 10 inches in diameter, the pitch being 21.5 feet. It is the largest cast steel propeller ever made in this country, its finished weight being 14,000 pounds. The diameter of the crank shaft and propeller shafts is 15 inches.

The *John Englis* is designed to make the run between New York City and Portland, Maine, in 20 hours, the distance being 320 nautical miles.

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YACHTS.

Margarita.—This twin screw steel yacht, built in Troon, Scotland, and described on page 660 of the August number, finished a series of progressive trials on August 6. The machinery worked well and there was no vibration of the vessel at either high or low speeds.

The speed under natural draft was 14 knots, and under forced draft, 16.5 knots. Runs were made over the Skelmorlie measured mile, from which the above speeds were obtained.

Sovereign.—This twin screw steel yacht, designed by Mr. J. Beavor-Webb of New York, and described on page 659 of the August number, tested her machinery and speed on August 28 by racing with and beating the Jersey Central Railroad Co.'s twin screw steamer *Monmouth*. The following account is condensed from "The New York Times."

The *Monmouth* left the Atlantic Highlands Pier at 8.30 A. M., notice of the coming race having been previously given by the owner of the yacht, Mr. Borden. The *Sovereign* was waiting at Buoy No. 10, and when the *Monmouth* drew near, started ahead. At the start the steamer had a clear lead, the yacht running up on the starboard side. The *Monmouth's* engines were making 144 revolutions per minute, but, as the yacht was slowly gaining, the revolutions were increased to 154 per minute. The *Monmouth* was then doing her best.

The *Sovereign* still gained. Off Swinburne Island the two boats were abreast. Inch by inch the *Sovereign* crept ahead, and when off Hoffman Island, her bowsprit was just ahead of the stem of the *Monmouth*. As they passed through the Narrows, the yacht was a quarter of a length ahead. At Quarantine it was half a length, and when abreast of Robbins Reef Light, the *Sovereign* was a clear length in front.

By the time Castle William, on Governors Island, was abeam, the yacht was several lengths in the lead, and at the Battery, was a quarter of a mile ahead.

Capt. Martin of the *Monmouth* said the steamer was badly handicapped by having so many passengers on board. They crowded to one side of the boat and gave her quite a list. He thought that if the *Monmouth* had not been so handicapped she would have won.

The *Monmouth* usually makes the trip in one hour and six minutes. On the day of the race, she made it from pier to pier in exactly one hour. At this rate, the speed would be about 17.25 knots.

———.—This twin screw steel yacht, building for Mr. C. R. Flint, New York, and designed by Mr. C. D. Mosher, New York, presents a novel feature in the removable deck house. This house is so constructed that it can be removed and replaced by a turtleback deck, with conning towers forward and aft, so that the yacht would be transformed into a torpedo boat.

The hull is subdivided by seven steel bulkheads, with a collision bulkhead forward. The length of the yacht is 122 feet over all; beam, 12.5 feet; draught, about 4 feet; guaranteed speed, 33 knots. The coal supply will be sufficient for a passage under steam across the Atlantic Ocean.

The vessel will be fitted for carrying two 14-inch torpedoes, four 6-pounder and four 1-pounder rapid fire guns.

The twin screw, quadruple expansion engines are designed for 2,000 I.H.P. There will be two Mosher's patent water tube boilers.

Scud.—This yacht was described on page 662 of the August number. The following particulars of the Blechynden water tube boiler are from "The Engineer," London.

The boiler is rated at 600 H.P. The diameter of the upper steam drum is 2 feet 2 inches, and its length 12.25 feet. The two lower water chambers are each 12 inches in diameter. The generating tubes are of galvanized steel, 1 inch in diameter; the four downcomer tubes, two at each end of the boiler, are 4.25 inches in diameter. The tube heating surface is 1,200 square

feet, and the grate surface, 24 square feet. The weight of the boiler complete, with water, smoke pipe and casings, is 6.25 tons.

The generating tubes can be cleaned internally, or withdrawn through holes in the top of the steam drum. These holes are closed by screw plugs, and are so placed that from forty to fifty tubes can be drawn through each hole. The generating tubes are expanded at each end, and are slightly curved, the longer side being nearer the furnace.

The illustration of the Blechynden boiler, given on plate VII, following page 368 of Volume VI, shows the general construction. The *Scud's* boiler, however, has no tubes on the circular part of the water chambers, and the tubes are all curved one way.

Speedy.—This twin screw steel yacht, launched on April 15, was partially described on page 434 of the May number. Many conflicting data have been published, but the following details are supposed to be correct. They are from "Engineering," London.

The designs are by Mr. Dixon Kemp, London. The "yacht measurement" of the *Speedy* is 140 tons, her length, 106 feet; beam, 16 feet, and depth, 11 feet 5 inches. She is capable of attaining a very fair rate of speed, 12½ knots having been maintained throughout a five hours' run with natural draft, while 13 knots were reached on the measured mile under similar conditions, the engines indicating 400 horse power. The engines are of the triple expansion type, and are fitted with direct acting steam reversing gear. The cylinders are 12, 19 and 30 inches in diameter, and the stroke, 21 inches. There is one single-ended boiler of the ordinary marine type, designed for a working pressure of 180 pounds per square inch. This boiler is 10 feet 9 inches in diameter and 9 feet long, the thickness of the shell plating being 1 inch. There are two furnaces, each 3 feet 4 inches in diameter, and separate combustion chambers. There are 304 tubes, 3 inches in outside diameter. The ordinary tubes are No. 8, B. W. G., and the stay tubes, ⅜ inch thick. The total grate surface is about 40 square feet. The shell consists of a single plate, so that circumferential joints are not required. The longitudinal joint is treble-riveted, every other rivet in the third row

being omitted, with inside and outside butt straps of the same thickness as the plate. The efficiency of this joint is very high, the plate section being 86.7 per cent. of the solid plate and the rivet section, 86.9 per cent. Further trials of the *Speedy* have given under forced draft a speed of 13½ knots, which could have been exceeded had the propeller been somewhat smaller, as the engines were unable to use all the steam supplied by the boiler. The yacht is fitted up very completely throughout, having steam windlass, electric light, distilling apparatus and a Daimler motor launch. The electric plant is run by a Parson's steam turbo-generator. The coal capacity is 28 tons and the water capacity, 3,000 gallons.

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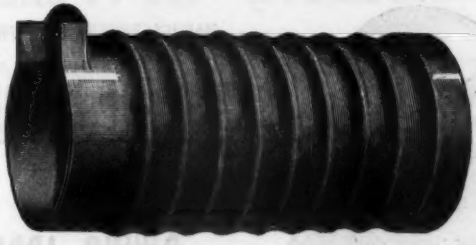

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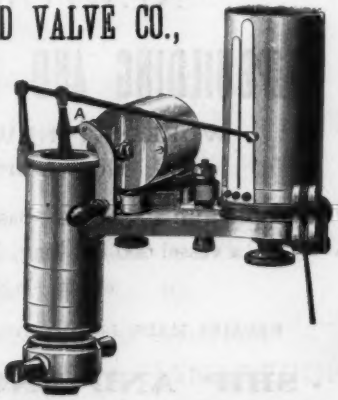

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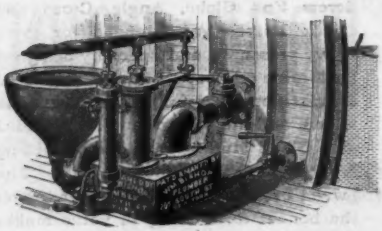

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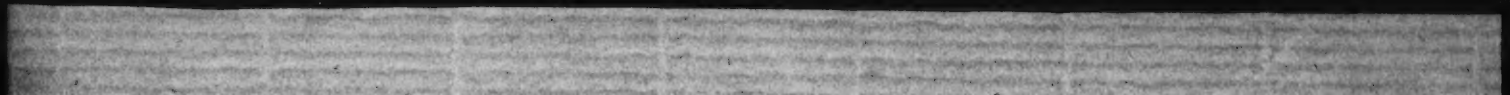
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